Reducing Cooling Energy Consumption in Data Centres and Critical Facilities

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MSc Built Environment: Environmental Design & Engineering

Gareth Cross

In particular I would like to thank the following for their great help in providing the raw data which form the basis of the calculations undertaken.

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Contents

1.0 Executive Summary

Section 6.0 presents all of the energy saving results in a uniform tabulated format to allow comparison.

The geographic location of the data centre is a significant factor when determining the energy savings available at either air cooled central refrigeration plant or outdoor condensers. An environment with an increased frequency of cold external ambient temperatures will result in increased plant efficiency.

An elevated chilled water temperature range increases the free cooling that may be achieved from air cooled central refrigeration plant.

Geographic location does not contribute towards the energy savings that can be achieved through either pump or fan variable speed drives.

Pump energy savings are generally only achievable when the data centre is not operating at peak cooling capacity. As a rule, this is during the data centre population period.

The use of elevated chilled water temperatures reduces the efficiency CRAC units in relation to absorbed fan power, by increasing the volume of air required to deliver a set amount of cooling.

In all instances analysed, a significant energy saving could be achieved through the utilisation of a plant or controls regime, that makes use of all of the installed plant capacity running at part load. This can be termed a 'hot standby' plant or controls arrangement.

All forms of energy saving should be considered where possible to reduce energy consumption and limit carbon emissions. However, each individual data centre needs to be assessed on its individual merits to determine the most relevant technologies.

2.0 Introduction

Given the rise of our everyday reliance on computers in all walks of life, from checking the train times to paying our credit card bills online, the need for computational power is ever increasing. Other than the ever-increasing performance of home Personal Computers (PC's) this reliance has given rise to a new phenomenon in the last 10 years ago. The data centre. Data centres contain vast arrays of IT cabinets loaded with servers that perform millions of computational equations every second. It is these data centres that allow us to continue with our reliance on the internet and the PC.

As more and more data centres become necessary due to the increase in computing processing power required for the everyday activities we all take for granted so the energy consumed by these data centres rises. Not only are more and more data centres being constructed daily, but operators are also looking at ways to squeeze more processing from their existing data centres. This in turn leads to greater heat outputs and therefore requires more cooling. Cooling data centres requires a sizeable energy input, indeed to many megawatts per data centre site.

Given the large amounts of money dependant on the successful operation of data centres, in particular for data centres operated by financial institutions, the onus is predominantly on ensuring the data centres operate with no technical glitches rather than in an energy conscious fashion.

This report aims to investigate the ways and means of reducing energy consumption within data centres without compromising the technology the data centres are designed to house. As well as discussing the individual merits of the technologies and their implementation technical calculations will be undertaken where necessary to determine the levels of energy saving, if any, from each proposal.

To enable comparison between each proposal any design calculations within this report will be undertaken against a notional data facility. This data facility will nominally be considered to require 1000 kW. Refer to Section 2.1 'Outline of Notional data Facility for Calculation Purposes' for details of the design conditions and constraints of the energy consumption calculations.

2.1 Outline of Notional Data Facility for Calculation Purposes

For the purpose of this report it is necessary to complete calculations to determine the actual level of energy saving, if any, by each proposal or item for discussion. Therefore a 1000 kW data facility will be used in each calculation to allow like for like comparison between the effectiveness of each individual option. In addition using the same size data facility will allow an approximate total energy saving to be calculated within the conclusions section without the necessity to use correction factors to multiply up or down the calculated energy savings.

Listed below are the design criteria for the notional data centre.

- The data centre will have a net sensible cooling duty of 1000 kW.
- The data centre will have a sensible / total heat ratio of 1.0.
- The data centre will operate 24 hours a day, 7 days a week and for 365 days of the year.
- The data centre will have '2N' redundancy for all outdoor central refrigeration plant (either air cooled chillers or outdoor condensers).
- The data centre will have '2N' redundancy for all indoor Computer Room Air Conditioning (CRAC) units.
- 'N' and '2N' plant areas to be separate to provide additional resilience.
- The data centre will be populated from day one to full population at the rates listed below.
    - 0 – 5 months (i.e. day 1 opening) = 20% of maximum data centre load
    - 6 – 11 months = 30% of maximum data centre load
    - 12 – 17 months = 40% of maximum data centre load
    - 18 – 23 months = 50% of maximum data centre load
    - 24 – 29 months = 60% of maximum data centre load

- 30 – 35 months = 70% maximum data centre load
- 36 – 41 months = 80% of maximum data centre load
- 42 – 47 months = 90% of maximum data centre load
- 48 months Onwards = 100% of maximum data centre load

The data centre build up from day one is due to the fact, that whilst the data centre will be fully operational on day one the IT team operating the data centre will not have had time before 'power on' to fully populate the data centre. Given the significant testing of IT server cabinets and equipment prior to turn on, this population can for a large data centre, take years as opposed to weeks. The author's experience working on such critical facilities, leads to the broad brush assumption that 48 months is a suitable period for full population.

Refer to Appendix 'Data Centre Layout' for scheme drawing of data hall and nominal cabinet layout.

Note: The tern '2N' redundancy refers to having 2 No. systems running in parallel but using diverse routes and plant space where possible, so in the event of one system failing the parallel system simply starts and takes over with minimal, if any, down time.

2.1.1 Cabinet Size and Configuration

As a rule standard cabinets loaded with blade servers pass air from front to back. The temperature difference (or ΔT) from the front to the back of the cabinet is normally in the order of 10 to 11°C at an air volume, set by the single speed blade server integral fan. Subjecting the components to temperature differences greater than this can induce what is termed 'thermal shock' and lead to component failure. A temperature difference below this generally will not cool the components sufficiently, leading to overheating and either automatic shutdown or failure.

The standard practice within data centres is to arrange the cabinets in what is termed a 'hot aisle / cold aisle arrangement'. This works by arranging cabinets so that they receive coolth via cooled air from a common cold aisle from the pressurised floor void via adjustable floor grilles and reject heat from the back of the cabinet into a common hot aisle. If there are large heat gains within the data centre it is often necessary to remove the heat from the hot aisle at source either via a plenum ceiling or via ductwork. This prevents the heated air contaminating the cold aisle and raising the on-cabinet temperature and therefore reducing the temperature difference across the cabinet, which in turn could lead to overheating. For a best practice solution hot and cold aisles are arranged alternately to make best us of the space.

All cabinets within the data hall will be nominally assumed to be 800 mm (wide) x 1000 mm (deep) x 47U (high).

1U, equals 1.75" (inches), is the standard measurement of measuring height for equipment cabinets. This property is used frequently within the IT industry.

Each hot aisle will be 1200 mm wide and each cold aisle will also be 1200 mm wide.

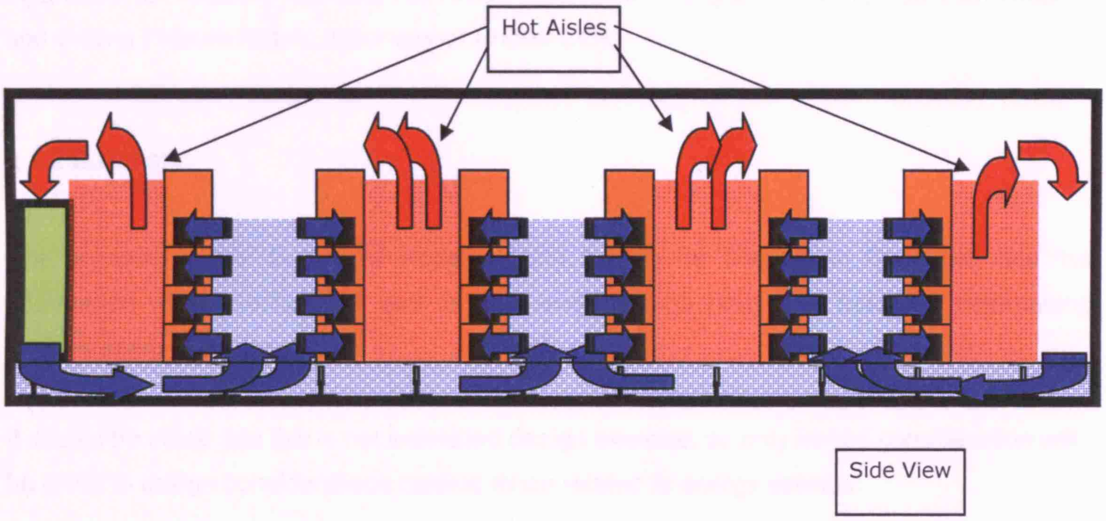


Figure 2.1.1a – Graphic Showing Hot Aisle / Cold Aisle Arrangement Using Downflow Units and Top Entry Air Return (Courtesy of Uniflair Ltd).

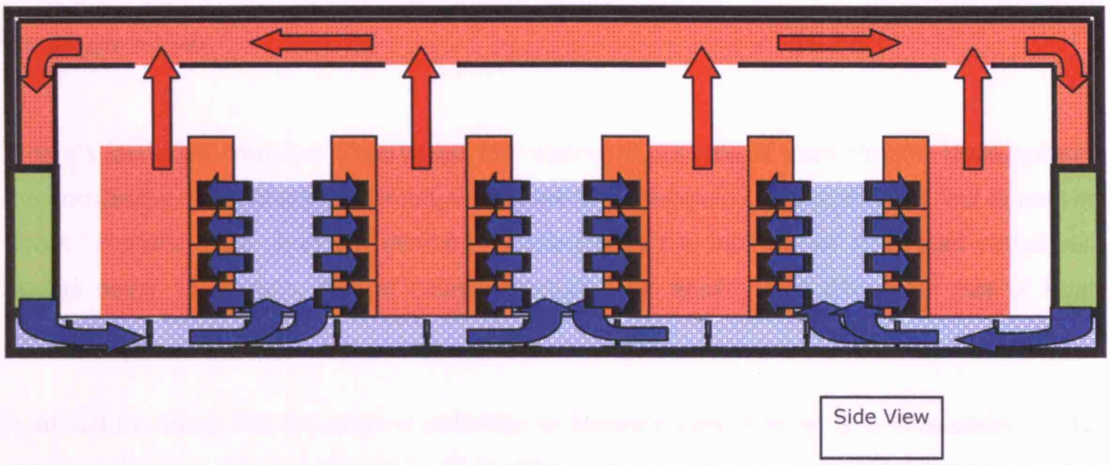


Figure 2.1.1b – Graphic Showing Hot Aisle / Cold Aisle Arrangement Using Downflow Units and Ceiling Plenum Return (Courtesy of Uniflair Ltd).

2.1.2 Summary

The purpose of the 'Outline of Notional Data Facility for Calculation Purposes' and the information within Section 2.1 and its sub-sections is to provide a basis for determining primary energy savings.

It should be noted that this is not a detailed design exercise, so only limited consideration will be given to design considerations outside those related to energy savings.

3.0 Moore's Law

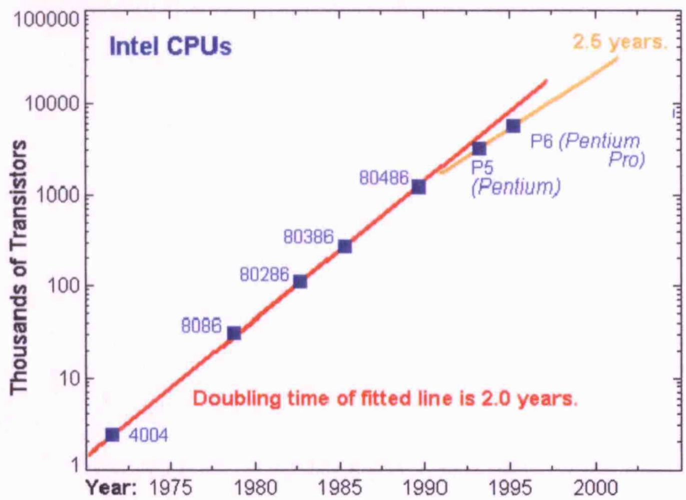
Moore's law is an empirical observation that states 'at our rate of technological development the complexity of an integrated circuit, with respect to minimum component cost, will double in about 18 months.' In short the numbers of transistors per inch on an integrated circuit will double every 18 months. It is of course the transistor density that drives the rate of heat generation.

It should be noted that the original definition of Moore's Law referred to a time period of 12 months. However this has slowed to 18 months over the last few years. Most experts expect Moore's Law to hold for at least the next decade. Gordon Earl Moore (born January 3rd 1929) is the co-founder of Intel Corporation. Moore's Law was originally published in an article in *Electronics Magazine* on the 19th April 1965.

It is Moore's Law that is continually driving the expansion of IT dependant industries. The requirement for additional computing power is a driving force behind the ever increasing number of data centre facilities being built, not just by large financial institutions and internet service providers, but by entrepreneurial data centre developers who sub-let processing power to smaller businesses. These data centres not only provide processing power to smaller businesses, but also act in some instances as a non-critical overspill facility for larger operators who would normally build and run their own critical facilities.

As a rule large financial institutions like to keep their data operations in-house to prevent any possible security breaches.

Figure 3.0a – Graphical Representation of Moore's Law



In the above graphic please note the slightly decreased rate in which transistors are 'multiplying' over the last few years as discussed above.

4.0 Resilience and Redundancy in Data Centres and Critical Facilities

The failure of the systems and servers within data centres can lead to the haemorrhaging of millions of pounds worth of business, in particular, when dealing with large financial institutions. Therefore the owners and operators of these facilities demand highly resilient systems to mitigate potential problems, and if at all possible, prevent system failure completely.

This resilience can take many forms. Dual risers and on floor ChW distribution systems, which provide multiple Chilled Water (ChW) paths to prevent total pipework failure, is a common method of increasing the resilience of the system. Dual ChW pipework would require a large number of actuator controlled changeover valves local to each CRAC unit. In a similar vein, if the indoor CRAC units were Direct Expansion (DX) as opposed to ChW, it would be good practice to utilise twin refrigerant coils within the indoor CRAC units coupled with 2 No condensers situated within separate outdoor plant locations.

However both of the above options come with a significant cost over and above the minimum necessary to provide the required cooling. Nevertheless, providing the minimum required services will not, as a rule, provide the most resilient solution.

Given that even the best Planned Preventative Maintenance (PPM) will not prevent downtime of plant on occasions, spare plant must be available to undertake the duty of the plant suffering from downtime. The 'spare' plant levels are known as redundancy. The level / size of plant (indoor or outdoor) required to perform the task to its design is termed the 'N' level of plant. Dependant on the nature of the data centre the plant levels can then be enhanced by a redundancy factor. These redundancy levels typically range from 'N+1' through 'N+25%' and 'N+50% to '2N'. Levels of redundancy above '2N are rarely seen.

Single Point of Failure (SPOF) analysis is generally undertaken to determine whether the levels of resilience and redundancy designed into the project are sufficient to ensure any one item of plant failure will not impact on the on-floor server operation

4.1 Run and Standby / Redundancy Plant Arrangements

Data centre operators must always consider how they intend to utilise their standby plant. This allows the designer to ensure the correct controls regime is provided. The traditional method of using standby plant was to allow it to remain idle, but primed, for operation whilst the 'N' level of plant operated at 100% of capacity. This is termed a 'run / standby' arrangement. With the advent of more complex controls capabilities, designers now more frequently use an arrangement that goes under many names, but for the purpose of this report will be termed a 'hot standby'. This arrangement utilises all of the installed plant but at a reduced capacity. For example, if the plant is installed in a '2N' arrangement then all of the plant would run at approximately 50% capacity, based on a constant load.

Benefits arise from using this arrangement when energy savings can be derived from running the plant at part load. This is often relevant in respect to free cooling plant in particular. For example, when the reduction in fan speed offers a greater reduction in absorbed fan power than the reduction in fan capacity. This is in line with the fan laws (i.e. RPM = $Power^3$).

This report will where possible highlight the benefits in energy savings achievable by using both arrangements

5.0 Proposed Methods of Reducing Energy Consumption in Data Centres (Mechanical)

Within this section four methods of energy reduction will be considered. Each option will be appraised and where possible an annual energy saving in both kilowatt-hours (kWh) and tonnes of carbon emitted (tonne.C) will be presented.

Note: The term mechanical energy will be used to describe energy consumed by mechanical plant such as chillers, fans and pumps. The scope of this report will not include energy consumed by the actual IT equipment.

The options that will be analysed are as follows.

- Option 01 – The incorporation of chillers with the potential to utilise 'free' cooling, in place of standard chillers that do not.

- Option 02 – The utilisation of a refrigerant based cooling solution which removes heat local to the cabinet by means of a evaporative coil.

- Option 03 – The use of Variable Speed Drives (VSD's) to determine the energy savings on both secondary and primary ChW pumps.

- Option 04 – The use of Variable Speed Drives (VSD's) to determine the energy savings possible on the fans within the CRAC units.

Assumptions will be stated within the relevant report section. Calculations and graphical output will be presented within the appendices, in addition relevant graphics and calculations will be presented within the main body of the text. All calculations have been undertaken using spreadsheets created by the author in Microsoft Excel (MS Office 2003 version). Where relevant, the methodology and processes contained within the calculation spreadsheets will be detailed within the main body of the text.

5.1 Free Cooling Chillers

The term 'mechanical cooling' can be used to describe cooling that is derived from an electrical input.

Recently, given the rise in CO_2 emissions and the United Kingdoms obligations under the Kyoto agreement there has been the necessity to reduce the amount of CO_2 released to atmosphere. Electricity production is responsible for a vast percentage of the United Kingdom's CO_2 emissions.

In turn the CO_2 output that is considered to be the responsibility of air conditioning buildings throughout the UK is also significant.

Therefore any measure that reduces the CO2 output from these sectors should be considered for discussion. One method of reducing the energy consumed for cooling of buildings and therefore electricity production is to utilise what are termed free cooling chillers.

Free cooling air-cooled chillers operate by using the outside temperate to cool the ChW as opposed to using the refrigeration cycle and a heat exchanger, as in a standard non-free cooling air-cooled chiller.

The term 'free cooling' can be considered a misnomer as the cooling derived is not entirely free of any electrical input. For instance there will always be an electrical load requirement to power the fans to draw air over the 'free cooling' coil. This fan electrical load is required irrespective of whether the air cooled chiller is operating in 'free cooling' or 'standard' mode. It would be more accurate to describe the term 'free cooling' as 'low energy cooling'. However given as this is an industry standard term, it will continue to be used within this report.

Assuming a fully functioning free cooling chiller has been installed the actual amount of free cooling achievable is dependant primarily on two factors; the chilled water temperature used within the space and the outside (external ambient) dry bulb temperature. Clearly, there is no scope for free cooling if the flow chilled water temperature is 8°C and the outside temperature is in excess of this. Therefore the lower the chilled water temperature range, the less scope for achieving any degree of free cooling. The higher temperatures required to achieve free cooling reduce the output of indoor units (within data centres, CRAC units) as the mean coil temperature is increased. This requires additional indoor CRAC units to meet the given load (i.e. notionally 10000 kW).

Air cooled chillers will also require a higher electrical input and possibly a larger foot print to produce water at the lower temperature, in addition, to more or larger CRAC units. Within data centres it is often a play off between the numbers of indoor units that can be accommodated within the plant space available, and the desire to achieve 'free cooling'.

Whilst lower water temperatures (for example the conventional ChW flow / return temperatures of 6 / 12°C) offer the highest kilowatt output from the indoor CRAC units, temperatures in this range are not conducive to achieving significant, all year round, free cooling. Given this low mean coil temperature, as the air passes over the coil, moisture condenses on the coil therefore there is a need to re-humidify the air. This action requires a significant energy input to evaporate water at high temperature within the air stream to humidify the air.

All of these factors have a bearing on the design of data centres and the potential system energy savings. This section of the report will investigate the parameters of achieving free cooling and the design considerations.

In addition the report looks at the bearing of location on the ability to achieve substantial free cooling. Clearly a data centre located in a cold region of the UK will utilise the more numerous and indeed severe bouts of cold to maximise the percentage of the overall cooling demand that is achieved without using the electrically powered mechanical refrigeration cycle.

To allow this analysis the free cooling has been analysed using the standard Chartered Institute of Building Services Engineers (CIBSE) Weather Files known as Test Reference Year (TRY) files. "The Test Reference Year consists of hourly data for twelve typical months, selected from approximately 20-year data sets (typically 1983-2004), and smoothed to provide a composite, but continuous, 1-year sequence of data. They enable the likely energy consumption of buildings to be assessed by simulation under typical weather conditions." Reference [2]. The TRY files are predominantly based on United Kingdom Meteorological Office (Met Office) data with short periods of missing data filled in using computer models and procedures designed for this purpose. The TRY files are considered to be the United Kingdom Building Services industry standard method of analysing energy consumption based on external weather conditions.

For the purpose of this report three separate TRY files have been used within the free cooling analysis, namely London, Manchester and Edinburgh. The author believes these three weather files offer a reasonable cover of different climates within the UK. In addition these areas are either the capital or a recognised regional hub and are therefore the most likely site of data facilities.

Listed below are the areas and their relevant 'official' CIBSE TRY file names.

- heb_try.wfl for London weather.
- ria_try.wfl for Manchester weather.
- tum_try.wfl for Edinburgh weather.

The calculation sheets 5.1.1a and 5.1.1b showing the spreadsheet calculations use the dry-bulb temperature annual frequency from the TRY files, to determine the amount of annual free cooling that will occur in a notional year, at the three locations (i.e. London, Manchester and Edinburgh).

Note: The external dry bulb ambient temperature needs to drop 2°C below the ChW flow temperature before any degree of free cooling is achieved. However, to achieve a significant degree of free cooling the temperature needs to drop further still.

5.1.1 Analysis for Chilled Water Flow and Return Temperatures 8 / 14°C

Based on the empirical data supplied by Stulz Gmbh for chilled water flow and return temperatures of 8 / 14°C the external ambient dry bulb temperature will need to drop below 6°C. This is shown in Figure 5.1.1a, which shows the percentage of free cooling achieved for any given external temperature down to -6°C. It can be seen that the percentage of free cooling increases steadily as the temperature decreases. The percentage of free cooling appears to increase significantly as the temperature drops below -3°C. The chiller as a standalone unit produces approximately 59% of its overall total cooling capacity (approximately 260 kW) as the temperature drops to -3°C without having to resort to using the internal mechanical refrigeration circuit and thus drawing substantial current (amperes) to run this refrigeration circuit. It should be noted, as previously discussed, that there is still a requirement for power to be supplied to run both the fans integral to the chiller, and to pump the fluid (ChW) around the cooling circuit.

Figure 5.1.1b - Graph Showing Annual Total Free Cooling Vs Temperature Based on ChW Flow / Return Temperature of 8 / 14 degC (Run / Standby Arrangement)

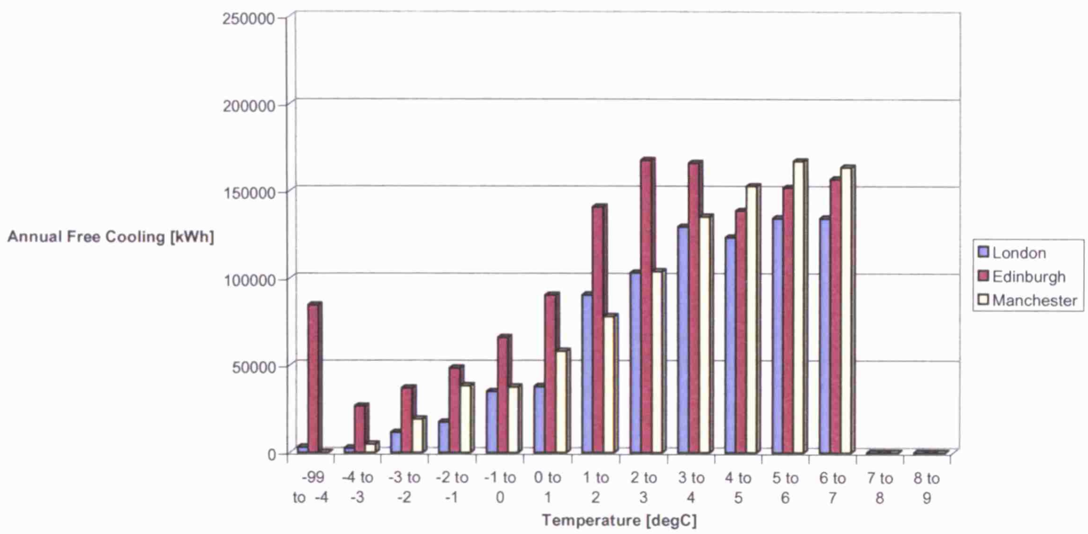


Figure 5.1.1b shows graphically the annual free cooling energy saving for the three separate geographical locations of London, Manchester and Edinburgh. The graph clearly shows that Edinburgh offers significantly more free cooling for each temperature band. This is evidently due to the increased number of annual hours that are spent in these temperature bands. This increased occurrence of hours when free cooling is achievable results in substantial energy savings over the course of the year. Manchester in general is second to Edinburgh in relation to free cooling achieved, with London and its relatively mild climate in third place. In fact in the temperature band of 4 to 6°C (inclusive) Manchester achieves greater free cooling due to the greater frequency of these temperatures than London.

As discussed in Section 4.1 'Run and Standby / Redundancy Plant Arrangements', within data facilities there is often the requirement for duplicate plant termed 'redundancy'. Therefore if there is a '2N' level of redundancy i.e. in the instance of central refrigeration plant there is 2 times the required amount of chiller plant, there will in effect be double the amount of chiller plant required to meet the total cooling load. This allows for the complete failure of one chiller circuit / system without compromising the data centre functioning.

The notional data centre for the purpose of calculations outlined within Section 2.1 requires 1000 kW of central refrigeration plant. The nominal chiller size used for all of the calculations within the appendices of this report is 250 kW; therefore 4 chillers are required to meet the total cooling load. However given the '2N' level of redundancy 8 No. chillers will be installed giving an installed capacity of 2000 kW.

The option then exists to either run 4 chillers at full load with 4 sitting idle. This operation method can be termed a 'run / standby' arrangement. In the event of one chiller failing it would then be necessary to start up a chiller that was standing idle. The next option is to run all eight chillers at 50% load. This arrangement can be termed a 'hot standby'. In the event of one chiller failing the other chillers would simply 'ramp up' to assume the additional load. This option has the benefit of not needing to engage the chiller start up cycle in the event of failure (a time costly process) which given the high heat gain within data facilities can lead to local over heating. The primary benefit of using the latter 'hot standby' arrangement is the ability to utilise the free cooling capacity of all 8 No. chillers.

Figure 5.1.1e - Graph Showing Total Cooling Required Vs Available Free Cooling from Chillers running in 4 No. Run / 4 No. Standby Arrangement (ChW Flow / Return Temperature of 8 / 14 degC)

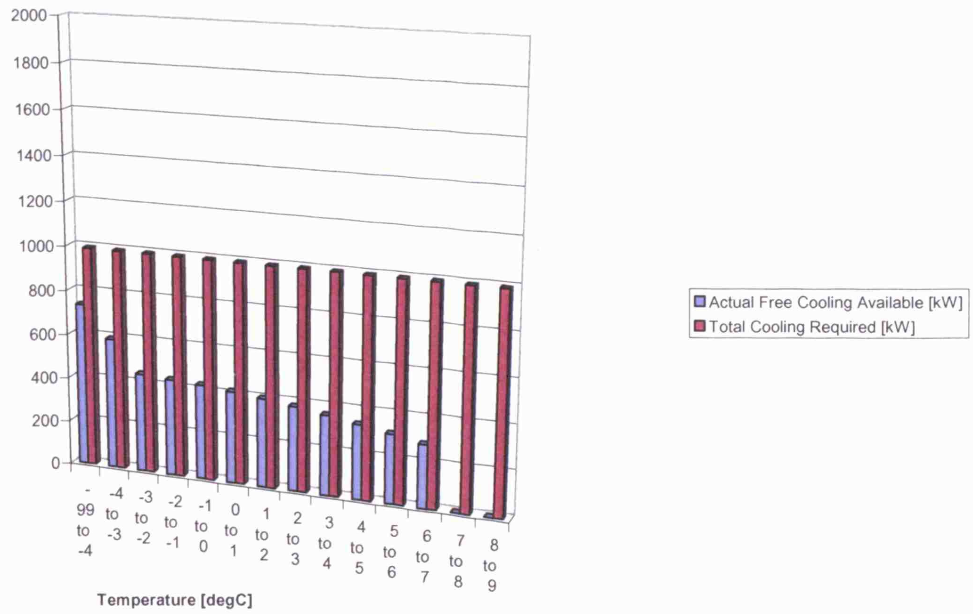
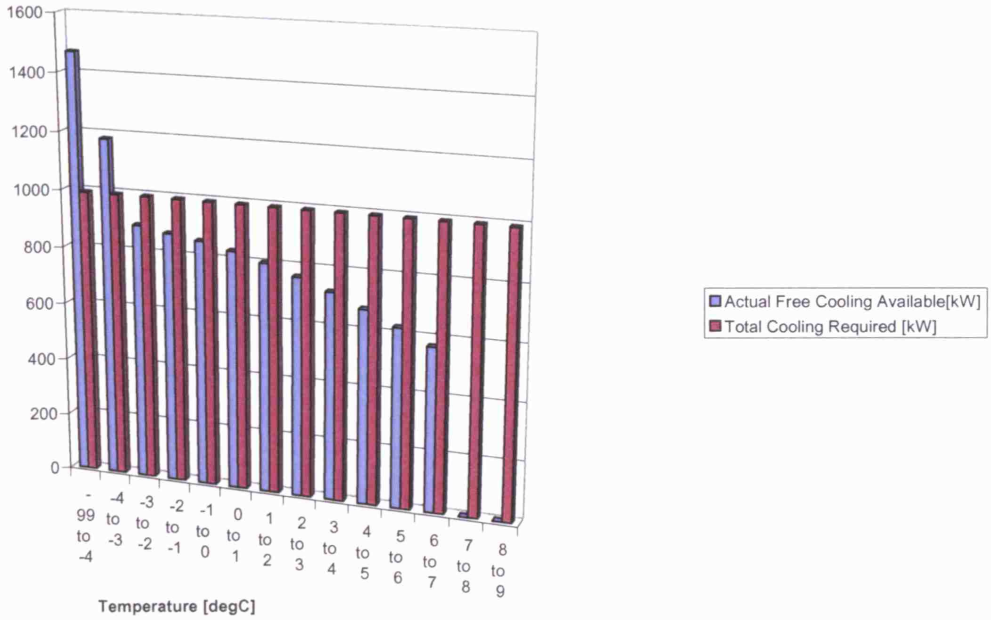


Figure 5.1.1f - Graph Showing Total Cooling Required Vs Available Free Cooling from 8 No. Chillers running in 'Hot Standby' Arrangement (ChW Flow / Return Temperature of 8 / 14 degC)



Figures 5.1.1e and 5.1.1f demonstrate that using the traditional 'run / standby' arrangement, as opposed to the 'hot standby' arrangement means that even in the coldest conditions four chillers can not meet the notional data centre full cooling load.

Figure 5.1.1e and Calculation Sheet 5.1.1a (see above and relevant appendices in rear of report) demonstrates that when the temperature is less than 4°C a maximum cooling duty of 734 kW (of a total cooling duty of 992 kW i.e. 74% of total cooling duty) can be achieved when a traditional 'run / standby' arrangement is employed.

Figure 5.1.1f and Calculation Sheet 5.1.1a (see relevant appendices) demonstrates that when the temperature is less than -4°C a maximum cooling duty of 1469 kW (of a total cooling duty of 992 kW i.e. 148% of total cooling duty) can be achieved when a 'hot standby' arrangement is employed. Even at 1°C, out of a total requirement of 992 kW, 800 kW of the total cooling load is provided via free cooling. Therefore it can be deemed highly advantageous to use a 'hot standby' arrangement and therefore maximise the free cooling available at any given temperature at which a condition exists that allows any proportion of free cooling.

The maximum free cooling available is only one half of the equation. This of course has to be coupled with the occurrence, or frequency of the temperatures required to achieve either complete or a percentage of free cooling. Figure 5.1.1c shows that London, Edinburgh and Manchester according to the calculations achieve an annual free cooling energy 'saving' of 822246 kWh, 1273915 kWh and 958632 kWh respectively at chilled water flow and return temperatures of 8 / 14°C when a traditional 'run / standby' central refrigeration (chiller) plant arrangement is used. This is based on a total annual energy requirement of 9408240 kWh. Therefore London achieves 9% of its total energy consumption via free cooling, Edinburgh 14% and Manchester 10%. Refer to Calculation Sheet 5.1.1a for calculations.

Figure 5.1.1d shows that London, Edinburgh and Manchester annually achieves an annual theoretical free cooling energy 'saving' of 1641832 kWh, 2484538 kWh and 1915761 kWh respectively at chilled water flow and return temperatures of 8 / 14°C when a 'hot standby' central refrigeration (chiller) plant arrangement is used. This is based on a total annual energy requirement of 9408240 kWh. Therefore London achieves 17% of its total energy consumption via free cooling, Edinburgh 26% and Manchester 20%. Refer to Calculation Sheet 5.1.1a for calculations.

5.1.2 Analysis of Chilled Water Flow and Return Temperatures 10 / 16°C

The ability to achieve a reasonable percentage of free cooling becomes a more viable option at the increased water temperatures.

Based on the empirical data supplied by Stulz Gmbh for chilled water flow and return temperatures of 10 / 16°C the external ambient dry bulb temperature will need to drop below 8°C for any significant free cooling to take place. This is shown in Figure 5.1.2a which shows the percentage of free cooling achieved for any given external temperature down to approximately -6°C. It can be seen that the percentage of free cooling increases steadily as the temperature decreases. The percentage of free cooling appears to increase significantly as the temperature drops below -3°C. The chiller as standalone unit produces approximately 79% of its overall total cooling capacity (approximately 260 kW) as the temperature drops to -3°C without having to resort to using the internal mechanical refrigeration circuit and thus drawing substantial current (Amperes) to run this refrigeration circuit. This compares with the figure of 59% when the chilled water temperatures are 10 / 16°C as opposed to 8 / 14°C. This can be explained as the chilled water temperatures are higher than external ambient thus delivering a greater percentage of free cooling.

Figure 5.1.2b - Graph Showing Annual Total Free Cooling Vs Temperature Based on ChW Flow / Return Temperature of 10 / 16 degC

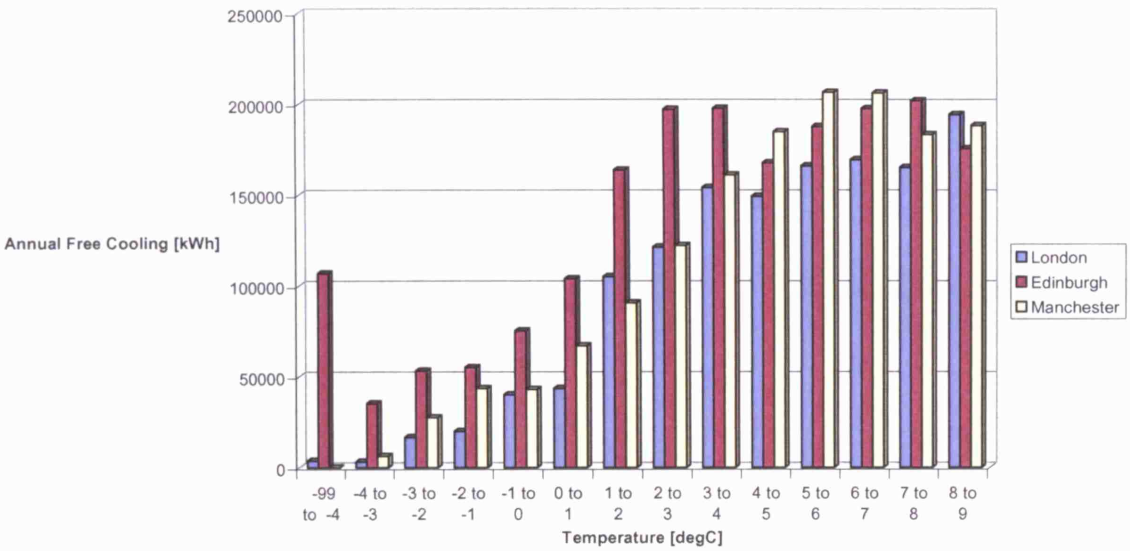


Figure 5.1.2b displays the same characteristics in relation to geographical location as Figure 5.1.1b, however given the increase in chilled water temperatures and therefore the increase in available free cooling the actual energy saving expressed in kWh is distinctly higher. Once again Edinburgh rates as the coldest followed by Manchester and then London meaning the greatest free cooling can be achieved in Edinburgh, and the least in London.

Using the same notional data centre conditions outlined within Section 2.1 coupled with the same levels of redundancy ('2N') and identical plant running arrangements (i.e. either 'run / standby' or 'hot standby') provide an accurate comparison between the varying chilled water temperatures allowing assessment of the actual energy savings on a like for like basis.

Once again it is demonstrated (via Figures 5.1.2e and 5.1.2f) that in the coldest conditions the traditional 'run / standby' chiller arrangement using four chillers as active with four further chillers in standby mode, cannot based on the empirical data meet the full cooling duty totally with free cooling. However as soon as the 'hot standby' arrangement is used sufficient free cooling is available at the coldest conditions. In fact at any temperature below -1°C almost all the entire cooling duty can be met by the free cooling available.

Figure 5.1.2e - Graph Showing Total Cooling Required Vs Available Free Cooling from Chillers running in 4 No. Run / 4 No. Standby Arrangement (ChW Flow / Return Temperature of 10 / 16 degC)

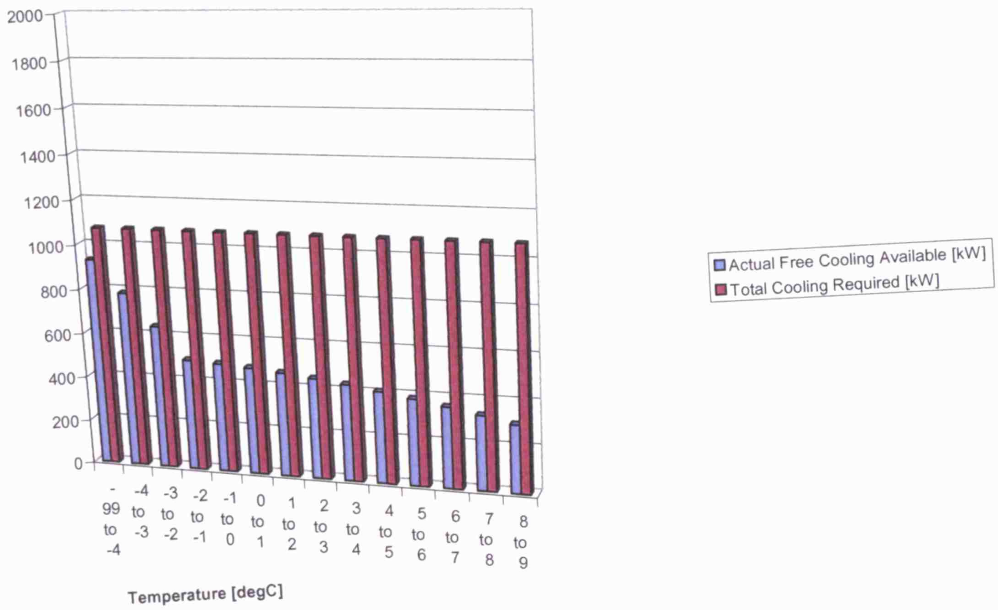


Figure 5.1.2e and Calculation Sheet 5.1.2a (see above and relevant appendices) demonstrates that when the temperature is less than -4°C a maximum cooling duty of 930 kW (of a total cooling duty of 1074 kW i.e. 87% of total cooling duty) when a traditional 'run / standby' arrangement is employed.

Figure 5.1.2f - Graph Showing Total Cooling Required Vs Available Free Cooling from 8 No. Chillers running in 'Hot Standby' Arrangement (ChW Flow / Return Temperature of 10 / 16 degC)

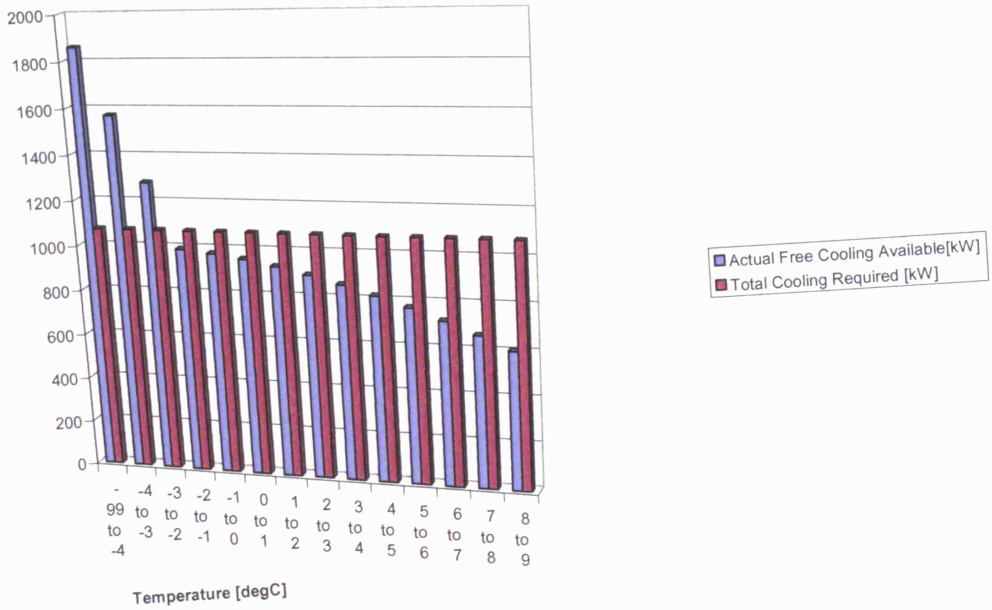


Figure 5.1.2f and Calculation Sheet 5.1.2a (see above and relevant appendices) demonstrates that when the temperature is less than 4°C a maximum cooling duty of 1859 kW (of a total cooling duty of 1074 kW i.e. 173% of total cooling duty) when a 'hot standby' arrangement is employed. Even at 1°C out of a total requirement of 1074 kW, 958 kW (89%) of the total cooling load is provided via free cooling. Therefore once again, it can be deemed highly advantageous to use a 'hot standby' arrangement and therefore maximise the free cooling available at any given temperature at which a condition exists that allows any proportion of free cooling.

Figure 5.1.2c shows that London, Edinburgh, and Manchester theoretically achieve an annual free cooling energy 'saving' of 1352378 kWh, 1919139 kWh and 1531188 kWh respectively at chilled water flow and return temperatures of 10 / 16°C when a traditional 'run / standby' central refrigeration (chiller) plant arrangement is used. This is based on a total annual energy requirement of 9408240 kWh. Therefore London achieves 14% of its total energy consumption via free cooling, Edinburgh 20% and Manchester 16%. Refer to Calculation Sheet 5.1.1a for calculations. This equates to an increase in percentage free cooling achieved of 5%, 6% and 6% for London, Edinburgh and Manchester respectively based on the increase in chilled water temperatures from 8 / 14°C to 10 / 16°C.

Figure 5.1.2d shows that London, Edinburgh and Manchester theoretically achieve an annual free cooling energy 'saving' of 2694219 kWh, 3708314 kWh and 3049458 kWh respectively at chilled water flow and return temperatures of 10 / 16°C when a 'hot standby' central refrigeration (chiller) plant arrangement is used. This is based on a total annual energy requirement of 9408240 kWh. Therefore London achieves 29% of its total energy consumption via free cooling, Edinburgh 39% and Manchester 32%. Refer to Calculation Sheet 5.1.1a for calculations. This equates to an increase in percentage free cooling achieved of 15%, 19% and 16% for London, Edinburgh and Manchester respectively based on the increase in chilled water temperatures from 8 / 14°C to 10 / 16°C.

5.2 Refrigerant Based Cooling Systems

There are a number of refrigerant based cooling systems on the market. These systems work by using the refrigeration cycle to remove heat from the cabinet or rack i.e. by evaporating the refrigerant at low pressure local to the components that require cooling. The refrigerant will then be condensed rejecting the heat either to another cooling medium or to the atmosphere.

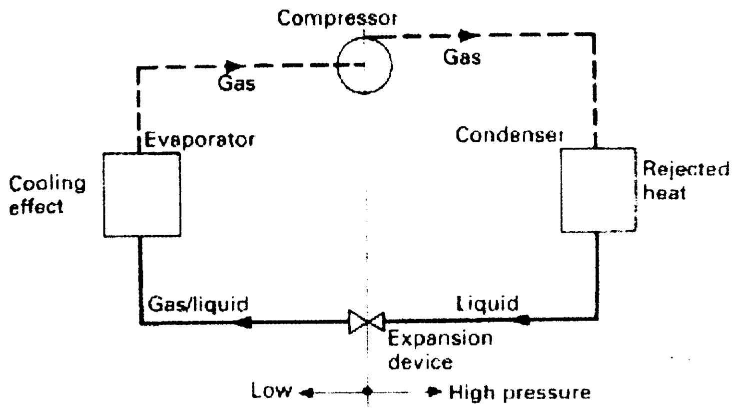


Figure 5.2a – Refrigeration Cycle [Reference 3]

Several manufacturers produce products that fulfil this role (including Knurr and Rittal), however for the purpose of this section of the report the Trox AITCS CO_2 will be discussed. Trox have stated that the CO_2 is capable of delivering upwards of 25 kW of cooling per 42U cabinet and in the region of 30 kW for the slightly larger 47U cabinet. Evidently this is far in excess of the 8 kW of cooling per cabinet, that can conventionally can be achieved [Reference 4] using a well designed air system. An air system is one that delivers cooling to the IT racks in the form of cooled air, however, this cooled air uses ChW as the primary cooling medium generated by air cooled central refrigeration plant. The cooled air is normally delivered via a pressurised floor void. This allows for a number of options, either a reduction in data centre foot print or more commonly data centres with higher loadings. Given that the air leaves the cabinet at the same temperature it enters there is no net heat increase to the space in which the cabinet is located. This means that refrigerant based systems lend themselves to retrospective inclusion in a space where there is no cooling capacity left as well as new build installations.

5.2.1 CO_2 as a Refrigerant

Refrigerant	**ODP**	**GWP**	**Comments**
R12	1	8500	Non-toxic and non-flammable, but very high ODP and high GWP.
R22	0.05	1700	Non-toxic and non-flammable, but very high ODP and high GWP.
R134a	0	1300	Non-toxic and non-flammable, no ODP but high GWP
NH3 (ammonia)	0	0	Highly toxic, non-flammable, no ODP and no GWP.
CO_2 (carbon dioxide)	0	1 (0)*	Non-toxic, non-flammable, no ODP and low / no GWP*

* a Global Warming Potential (GWP) of 0 can be achieved using industry reclaimed carbon dioxide. [Reference 9]

Table 5.2.1.a – Table of Refrigerant Properties

Ozone Depletion Potential (ODP) uses R12 as the base measurement of 1; therefore 1 is considered highly damaging to the ozone layer and 0 is considered to have no detrimental effect on the ozone layer.

Global Warming Potential (GWP) uses Carbon Dioxide (CO_2) as the base measurement assigning it once again a value of 1. However, most refrigerants have a higher Global Warming Potential than CO_2 and therefore the values will be higher than 1.

* "Although there are a number of ways of measuring the strength of different greenhouse gases in the atmosphere, the Global Warming Potential (GWP) is perhaps the most useful, particularly as a policy instrument. GWPs measure the influence greenhouse gases have on the natural greenhouse effect, including the ability of greenhouse gas molecules to absorb or trap heat and the length of time greenhouse gas molecules remain in the atmosphere before being removed or broken down, the atmospheric lifetime. In this way, the contribution that each greenhouse gas has towards global warming can be assessed." [Reference 5].

CO_2 as a refrigerant is also electrically benign (i.e. it does not react with electricity to cause fire) therefore lending itself to use in an environment such as a critical facility where it will be non-hazardous to equipment, servers and cabling. In effect, data centre users are hostile to using water as a heat transfer medium due to its reactive nature with electricity. This has forced critical facilities to consider the use of benign of heat transfer mediums and refrigerants such as CO_2.

Within the Trox CO_2 system the CO_2 remains at the same temperature throughout the system but changes phase under different pressures. It is during this phase change that the cooling effect is created.

The "latent phase change of CO_2 is 182 kJ/kg" [Reference 6]. This means that 1 kg of CO_2 can absorb 182 kJ of energy as it changes phase from liquid to gas.

The specific heat capacity of water at constant pressure is conventionally given as 4.18 kJ/kg.K. Assuming a standard temperature difference as used in most chilled water systems of 6k this results in 1 kg of water being able to absorb 25.08 kJ of energy. i.e.

$$4.18kJ / kg.K \times 6K = 25.08kJ$$

This means that using the phase change of CO_2 is 7.26 more effective than using water with a 6K temperature distance as a heat transfer medium. i.e.

$$\frac{182kJ / kg}{25.08kJ / kg} = 7.26$$

Note: The 6K temperature difference proposed above is an industry standard temperature difference. The above figure of 7.26 would be halved in the instance of using a temperature difference of 12K. However this would require an increased electrical input to the chiller to meet the increased chiller load.

It should however be noted that this is only a measure of the effectiveness of the heat transfer medium and not of the system efficiency. For instance this calculation does not take into account absorbed power associated with either compressors, or condensers in relation to the refrigerant cycle or pumps in relation to the ChW circuit.

The diagram below shows the Pressure / Temperature phases of CO_2 in a graphical format (often known as a P-T diagram).

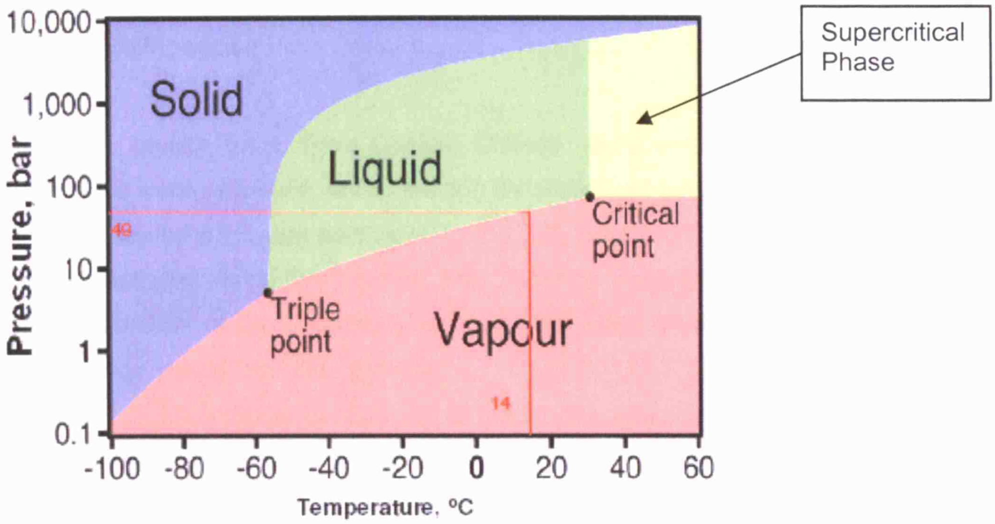


Figure 5.2.1e - CO_2 P-T Diagram [Reference 6, courtesy of Trox AITCS]

The diagram shows the three possible phases of CO_2, namely solid, liquid and vapour. "The triple point sets the lower temperature limit (-56.6°C) for the heat transfer process (to take place) based on evaporation and condensation, where the CO_2 coexists as all of its three phases. CO_2 cannot be liquefied above the critical point (31°C) which is the upper limit for usage as a volatile heat transfer fluid. The CO_2 system is charged to a pressure of 49 bar (A), which hold the CO_2 at a temperature of 14°C and above room dew point to avoid condensation. " [Reference 6].

The upper limit of 31°C the liquid and gas phases cannot exist as separate phases and is termed the "superfluid or supercritical fluid has properties indistinguishable from the liquid and gaseous phases." [Reference 7]. This renders the fluid unsuitable for use within the evaporation and compression refrigeration cycle.

As shown within the P-T diagram below, at the lower temperature limit of (-56.6°C) the fluid will change between the solid and vapour phases without passing through the liquid phase. "Physically, this boundary implies that the gas and solid can co-exist and transform back and forth without the presence of liquid as an intermediate phase. A solid evaporating directly into the gas is called sublimation. At normal atmospheric pressure and temperature, the stable carbon dioxide phase is gas. This means that the final product is gaseous carbon dioxide and this final state is independent of the initial phase (be that liquid, solid or gaseous). Any solid CO2 will just sublime" Reference [7].

The above description indicates why CO_2 is potentially a useful refrigerant. The one possible drawback is the high pressures required to maintain the CO_2 within the limiting boundaries.

5.2.2 Central Refrigeration Plant Performance – Trox / Star Refrigeration CPA500

Given that in section 5.1.1 'Free Cooling Chillers' air cooled refrigeration plant potential energy savings were assessed, in this section air cooled refrigeration plant energy savings as assessed in Section 5.1.1, are compared to the CO_2 refrigeration plant potential savings that have been assessed within this section. This allows a like for like comparison between the energy consumption of the 2 methods on a per kW basis within the conclusion section. All calculations are based on data provided by Trox AITCS / Star Refrigeration Ltd for the CO_2OLairpac CPA500 model i.e. 500 kW of outdoor central refrigeration plant. This machine uses CO_2 for the secondary refrigeration circuit refrigerant whilst using R134a (high GWP) for the primary side refrigerant. The CPA500 has a nominal cooling capacity of 500 kW.

Refer to Schematic 5.2.2a for a schematic representation of the CPA500 unit and 5.2.2b for a cutaway of the actual central refrigeration plant.

Calculation Sheet 5.2.1a - Trox AITCS / Star Refrigeration Data Showing COSP for 500 kW CPA500 Unit

	Load	External Ambient Temp							
	%	35 degC	30 degC	25 degC	20 degC	15 degC	10 degC	5 degC	0 degC
COSP	100%	3.18	3.67	4.30	5.01	5.83	6.48	7.91	7.91
	90%	3.33	3.89	4.53	5.27	6.14	6.81	8.02	8.58
	80%	3.37	3.95	4.62	5.42	6.32	7.05	8.13	8.78
	70%	3.30	3.94	4.68	5.53	6.52	7.43	8.25	9.01
	60%	3.13	3.85	4.64	5.55	6.70	7.70	8.96	8.96
	50%	2.96	3.69	4.52	5.54	6.74	7.83	8.83	8.83
	40%	3.03	3.66	4.59	5.61	6.88	8.76	8.76	8.90
	30%	3.61	4.37	5.20	6.16	7.26	8.36	8.52	8.52
	20%	3.06	3.79	4.52	5.34	6.41	8.18	8.18	8.18
	10%	1.00	1.20	1.50	3.70	5.70	5.68	5.68	5.68

Calculation Sheet 5.2.1a above shows the range of Coefficient's of System Performance at a range of different temperatures (0°C to 35°C in relation to heat rejection) against different percentages of the full load of 500kW (i.e. 50% of load is equal to 250 kW). Calculation sheet 5.2.1a clearly shows that there is a marked advantage in using the CPA500 machine at a reduced load in an environment where the temperature is minimal. COSP's of 9.0 can be achieved when the temperature is in the region of 0 to 5°C at loads ranging from approx 40% to 50% of capacity (200 to 250kW).

Figure 5.2.1a - Standard 500kW Air Cooled CO2OL Pack

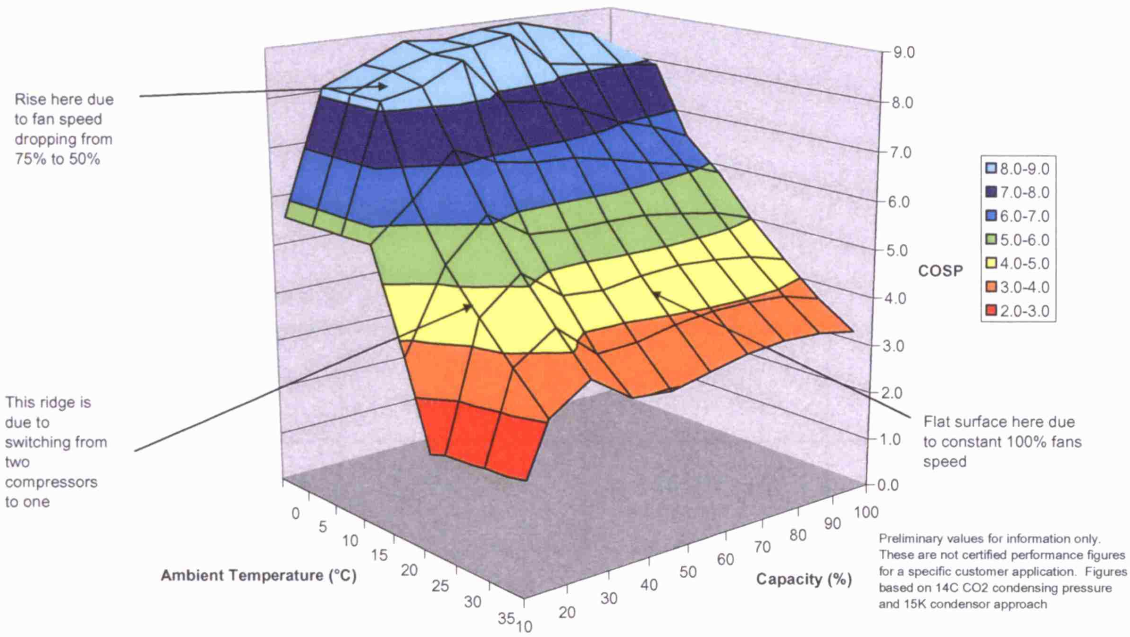


Figure 5.2.1a shown above uses the information in Calculation Sheet 5.2.1 also shown above to create a 3D graphical representation, taking into account three variables, namely external ambient temperature, % plant operating capacity and Coefficient of System Performance (COSP). In addition comment has been made showing why certain changes appear within the COSP given certain plant characteristics and conditions.

This peak performance at part load lead to the investigation of the possibility of using 4 chillers operating at part load (50%) as opposed to the 2 chillers operating at full load (100%) that would normally be required to meet the theoretical 1000 kW data centre load. As per the free cooling chiller calculations detailed in Section 5.1.1 'Free Cooling Chillers' this would mean the difference between running a traditional 'run / standby' arrangement and a 'hot standby' arrangement. Both control methods have been assessed within the calculations.

This was once again investigated using the London, Edinburgh and Manchester weather files to determine the best location or siting a data centre using the Trox AITCS CO_2 refrigerant system.

It should be noted that there will be a degree of inaccuracy within the presented calculation sheets due to the fact the CIBSE weather files are presented in 1 degree temperature bands whilst the Trox AITCS / Star Refrigeration data was presented in 5 degree temperature bands. This has been overcome by making the assumption that the refrigeration plant performs at the same COSP for all of the hours within that 5 degree temperature band i.e. the sum of all the hours within that temperature band.

Below is a tabulated summary of the findings of the following calculation sheets and the location of the sites .

Calculation Sheet 5.2.1.b – London (hebtry.wfl)

Calculation Sheet 5.2.1.c – Edinburgh (tumtry.wfl)

Calculation Sheet 5.2.1.d – Manchester (riatry.wfl)

Table 5.2.2a – Tabulated Results (kWh + Approximate Cost)

	London (hebtry.wfl)	Edinburgh (tumrty.wfl)	Manchester (riatry.wfl)
Annual Energy Consumed – 2 No. Chillers @ 100%	1299368 kWh	1226097 kWh	1258405 kWh
Annual Energy Consumed – 4 No. Chillers @ 50%	1132727 kWh	1067721 kWh	1092884 kWh
Annual Energy Saving	**166641 kWh**	**158370 kWh**	**165521 kWh**
Assumed Cost of Electricity	£0.07 per kWh	£0.07 per kWh	£0.07 per kWh
Approximate Annual Cost Saving	**£11,666.87**	**£11,085.90**	**£11,586.47**

Tables 5.2.2a shows the annual energy savings and the related cost saving. A CO_2 system located in Edinburgh costs less than an equivalent system in either London, or Manchester in both financial and energy terms. However the difference between the highest energy usage and the lowest energy usage (i.e. London and Edinburgh) is a mere 8271 kWh. This equates to a financial cost saving of £580.97 per annum.

Table 5.2.2b – Tabulated Results (tonne.C)

	London (hebtry.wfl)	Edinburgh (tumtry.wfl)	Manchester (riatry.wfl)
Annual Carbon Output – 2 No. Chillers @ 100%	152 tonne.C	143 tonne.C	147 tonne.C
Annual Carbon Output – 4 No. Chillers @ 50%	133 tonne.C	125 tonne.C	128 tonne.C
Annual Carbon Output Saving	**19 tonne.C**	**18 tonne.C**	**19 tonne.C**

Table 5.2.2b shows the annual tonnes of carbon emitted per year for all three geographical sites for both a traditional 'run / standby' plant arrangement and a 'hot standby' central plant arrangement. It can be seen that Edinburgh has the lowest annual carbon output saving, changing a traditional 'run / standby' plant arrangement to a 'hot standby'. This is despite it having the lowest annual carbon output of all three locations in the energy efficient 'hot standby' arrangement. This fact, coupled with the marginal difference in annual energy consumption shown across the three notional sites in Table 5.2.2a, leads to the conclusion that a CO_2 cooling system is less dependant on the external ambient temperature than a free cooling chiller using ChW as a heat transfer medium.

5.2.3 Discussion of Notional data Centre Design using Refrigerant Based System

Based on the assumed load of 1000 kW for the notional data centre it would require 50 x 42U cabinets CO_2OLrac (internal heat removal plant) units to remove the full 1000 kW load. This is based on each 42U cabinet being able to absorb 20 kW.

Each CO_2OLrac unit contains 5 No. fans installed on an 'N+1' basis, allowing there to be sufficient fan capacity in the event of one fan failing. This means that the 'N' capacity of fans is 4. Each of the fans has an approximate maximum absorbed energy of 140W or 0.140 kW. This means at maximum fan capacity each 42U CO_2OLrac unit absorbs approximately 0.56kW. Considering there is a total of 50 x 42U CO_2OLrac units this equates to a maximum absorbed fan power equal of 28kW.

However it should be noted that this figure will be significantly lower as in most instances the fans will not be operating at maximum capacity. Given that as fan speed drops the power absorbed drops proportionally to the inverse cube of the speed, this figure is likely to be significantly lower. Without undertaking detailed load analysis for a particular space, determining the actual fan power absorbed is not possible. For the purpose of this exercise, the figure can be deemed unquantifiable.

Assuming a worst case absorbed fan power of 28 kW constant load this equates to an annual energy consumed figure of 245280 kWh which in turn would cost an approximate figure of £17,169.60 (based on an electricity cost of £0.07 per kWh). Once again, please note this figure is likely to be significantly less in reality.

5.3 Variable Speed Chilled Water Pumping

There are 2 main ways of controlling the pump that drives the water flow within the ChW circuit. These are single speed pumping and variable pumping speed dependant on the pump load.

Single speed pumping, as the name suggests, utilises a single speed pump used in conjunction with 3-port diverting valve. Variable Speed pumping requires the pump speed to vary in conjunction with the cooling unit load. Variable speed pumping is primarily used in conjunction with 2-port valves.

A variable speed pump must of course be sized to deal with the full design load, however if this design load will not be required on a regular basis significant energy savings can be achieved. This is illustrated by the third of the widely accepted pump laws shown below. In addition the overall size of the pump can often be reduced, if a diversity factor is known relating to the load profiles of the system. In the instance of data centres and assuming a constant load this will not be considered in the context of this report.

$$\frac{P_2}{P_1} \propto \left(\frac{S_2}{S_1}\right)^3$$

Where P = Pump Power Absorbed
S = Pump Speed.

Given that the relationship between pump power absorbed, and pump speed is non-linear, substantial energy savings can be achieved by running the pump at speeds that are a reduction on the speeds required to run the pump at full design load.

"However, the calculation of pump power reduction should also take into account possible changes in pump efficiency. Pump power is related to pressure loss by the following equation:" Reference [1].

$$P = \frac{\Delta p.Q}{\eta}$$

Where Δp = Pressure Loss
Q = Volume Flow Rate
η = Pump Efficiency

5.3.1 Description of 2-port Control Valve System Operation.

The most common method of controlling the flow to individual indoor units, (be they heating or cooling coils) in this age of carbon consciousness, is to use a combination of 2-port control valves and differential pressure valves.

In the instance of the data centre, the 2 port valves are mounted in line on the flow pipework to regulate the ChW flow. These valves are generally controlled by actuators linked to the central controls system. When a demand exists for cooling the actuator, will open the valve to allow an increased flow of chilled water, which will in turn, via the air passed over the ChW coil, provide cooling to the data facility.

In addition to a 2 port constant flow regulator a Differential Pressure Valve (DPV) is required. These can be used on either the ChW sub-branches or can be used on each individual indoor unit. A DPV's purpose is to "maintain a constant pressure differential across a sub-branch (or indoor unit), thereby protecting the downstream control valves from excessive pressures, whilst nullifying the effects of pressure variations caused by the movement of control valves in other parts of the system." Reference [1].

In the instance of data centre design a DPV is normally installed on each indoor CRAC unit to ensure guaranteed control. This is normally cost prohibitive, however, in data centre design the capital cost required is generally judged to be a worthwhile expenditure due to the additional degree of protection this affords the CRAC unit.

Figure 5.3.1c and Figure 5.3.1d shown below show separate simplified schematics of both the individual CRAC unit DPV and sub branch DPV.

Some valve manufacturers manufacture a combined 2 port control valve and DPV.

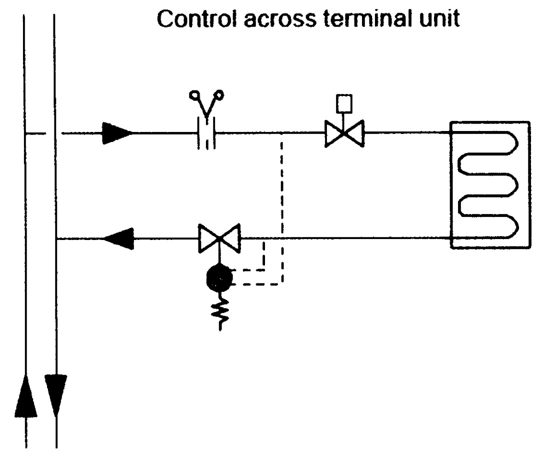


Figure 5.3.1c – Differential Pressure Control across a CRAC Unit [Reference 1]

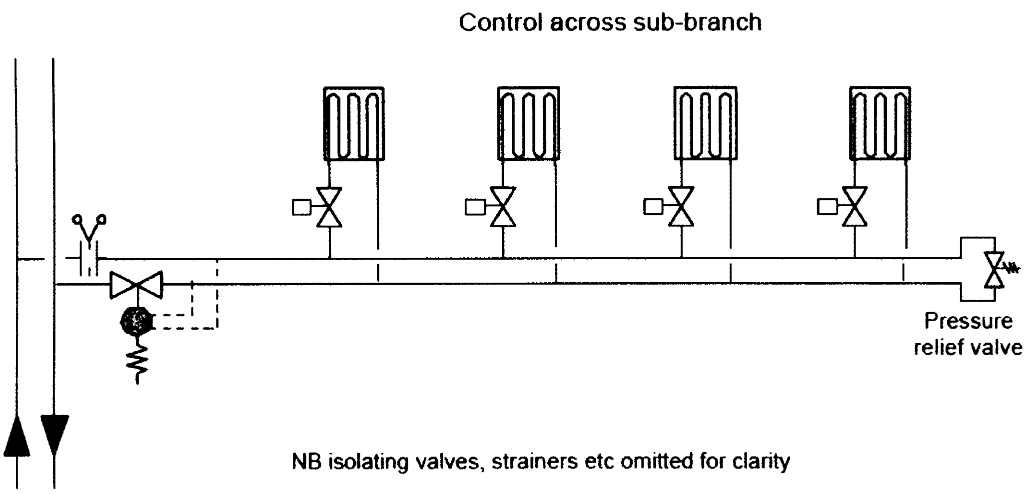


Figure 5.3.1d – Sub Branch Differential Pressure Control [Reference 1]

The installation of differential pressure valves means that the pump can work to a fixed pressure drop. Therefore the pump automatically responds to the cooling demand within the data centre. This, coupled with the previously discussed non-linear relationship between a reduction in pump speed and power absorbed, can offer generous energy savings over a given time period.

An additional benefit of using 2 port control and variable speed pumping is that by controlling by this method maintains a fairly constant ChW temperature differential (ΔT). This in turn assists with maximising chiller efficiency, given that the chiller load remains in the main, constant.

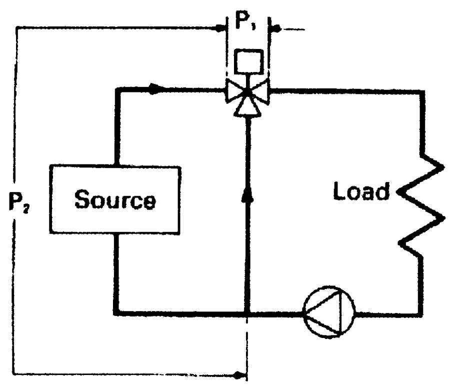
The topic of control of variable speed pumping circuits is a highly detailed subject. A full discussion and analysis is considered to be outside of the scope of this report. The above brief description is intended to be an introduction to the topic as opposed to an in-depth discussion.

5.3.2 Description of 3-port Control Valve System Operation.

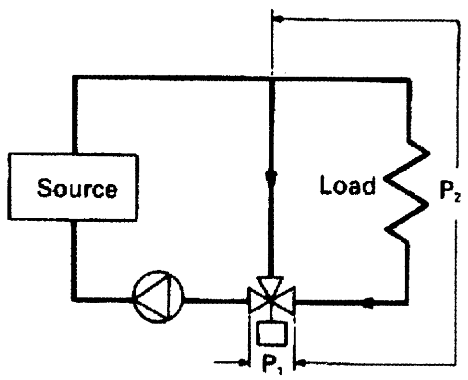
A single speed pump operates at one speed only. This pumping arrangement is therefore best suited to applications with a constant load.

A varying load at a CRAC unit could require a 3-port valve used in a mixing configuration i.e. with two inlets and 1 outlet. Using 3-port control valves in the diverting mode is not recommended, as not all 3-port valves "have the required internal pattern" that lends itself to this application. Reference [3].

Refer Figure 5.3.2b below details the 2 separate 3-port valve configurations



(a) 3-port mixing



(b) 3-port diverting

Figure 5.3.2b – 3-port Mixing Valve Schematics [Reference 3]

The figures above highlight how a 3-port valve based system controls the CRAC units.

However if the load varies by dropping off, therefore requiring cooling within the space to be reduced, the chilled water return temperature returning to the chillers will be higher than if the constant design load is maintained. The temperature differential (ΔT) will therefore vary, resulting in a chiller load that fluctuates; this will have an adverse effect on chiller efficiency and Coefficient of Performance (CoP).

5.3.3 Energy Saving in Period Prior to Data Centre Full Population

To simplify these calculations a single primary pump has been considered along with a single secondary pump. In reality each of these pumps would be part of a run standby arrangement as a minimum, and the run load would probably be spread across a number of pumps.

Based on a standard Grundfos pump selection, it is assumed that the same size pump will be used for both primary and secondary. In reality, the likelihood is that a number of separate run and standby pumps would be used to spread the risk of system failure.

It has been assumed that the data centre population is in line with that outlined previously in Section 2.1 'Outline of Notional data Facility for Calculation Purposes'. The data centre population has been stepped in incremental increases in cooling load for the purpose of calculation. However, in reality, this increase would be more of a linear increase, as IT staff introduce individual pieces of equipment on a one-by-one basis.

Based on the run currents provided by the manufacturer's selection the calculation below has been used to calculate the electrical kilowatt loading (K_e)

$$K_e = \sqrt{ph} \times V_{line} \times I_{line} \times pf$$

Where ph = Number of phases
V_{line} = Line Voltage (Volts)
I_{line} = Line Current (amperes)
pf = Power Factor (assumed to be 0.9)

The calculated K_e load of each of the 2 pumps is equal to 22kW each.

Within Calculation sheets 5.3.1a and 5.3.1b the kW loading of the pump considered is converted to energy consumption based on the stepped incremental data centre population and the pump laws discussed earlier within this section.

Calculation 5.3.1a shows pump power remaining steady from months 0 – 48 due to the single speed nature of the pump. This is in spite of the cooling requirement increasing throughout this period starting from 20%. Due to the low load within the space most of the chilled water pumped to the terminal CRAC unit will be diverted back via a 3 port mixing valve to the return pipework, rather than to the return via the cooling coil within the CRAC unit. This will result in the ChW temperature returning to the chiller only marginally reduced from the flow temperature i.e. a small ΔT. This will result in reduced chiller efficiency, this does not form part of the discussion within this section.

Calculation sheet 5.3.1b shows the energy saved in relation to pumping power given the usage of a variable speed pumping system. This calculation sheet shows that in the 48 month full population period the halfway point of full pump duty of 22 kW i.e. 11 kW is only exceeded after the 36 month. In fact over 26000 kWh were saved in 6 months of operation between operating at 100% to 90% (the last 6 months before the data centre was fully populated).

Calculation 5.3.1c - Chilled Water Pumping Energy Consumed Calculation Comparison - Comparison Sheet

Based on Theoretical Stepped Population of Data Centre - See Section 2.1

Months	Data Centre IT Populated	Single Speed Pumping Energy Consumed	Variable Speed Pumping Energy Consumed	Energy Consumed Percentage Reduction - See Note 1
	%	kWh	kWh	%
0 - 5	20%	96345	771	99%
6 - 11	30%	96345	2601	97%
12 - 17	40%	96345	6166	94%
18 - 23	50%	96345	12043	88%
24 - 29	60%	96345	20811	78%
30 - 35	70%	96345	33046	66%
36 - 41	80%	96345	49329	49%
42 - 47	90%	96345	70236	27%
48 onwards	100%	96345	96345	0%
		867108	**291348**	**66%**
	CO2 Output	**144.03 tonnes**	**48.39 tonnes**	

	Primary Pump	Secondary Pump	Total
Total Energy Saved Using VSD =	**575760 kWh**	**575760 kWh**	**1151519 kWh**
Total Cost Saved Using VSD=	**£ 40303.18**	**£ 40303.18**	**£ 80606.35**
Carbon Emission Indices (kg.C/kWh) =	**0.117**	**0.117**	**0.117**
Total Carbon Emission Saving (kg.C) =	**67364 kg.C**	**67364 kg.C**	**134728 kg.C**
Total Carbon Emission Saving (tonne.C) =	**67.36 tonnes**	**67.36 tonnes**	**134.73 tonnes**

Notes
1) The energy consumption reduction percenrage figure is for a given time period i.e. 0 - 5 months.
2) Based on Electricity Cost of £0.07 per kWh
3) Based on Carbon outputof 0.117 kg.C / kWh

Calculation sheet 5.3.1c details the comparison between the 2 pumping methods. It can be seen that between months 0 – 5 i.e. the first 6 months, the energy saving using variable speed pumping is equal to 99%, even between months 42 and 47 i.e. the last 6 months ,before full population, the energy saving is 27%. Over the total population period of 4 years this is approximately 66%.

This is equivalent to an energy saving of 575760 kWh over 4 years for either of the single or the primary pumps, which is based on 0.1661 kg CO_2 / kWh (for electricity production but ignoring transmission losses) equates to 95.63 metric tonnes of CO_2 over the initial 4 year population period.

Assuming, as previously described, that both the primary and secondary pumps are equal in size using a header arrangement there is a 4 year energy saving of 1151519 kWh. This equates to a 4 year saving in CO_2 output to the atmosphere of 191.27 metric tonnes.

Based on an energy production price of £0.07 this equates to an energy saving of £80,606.35 over 4 years.

Figure 5.3.1a - Graph Showing Energy Consumption Comparison of Single Speed Vs Variable Speed Pumping for Stepped Population of Data Centre (0 - 48 Months)

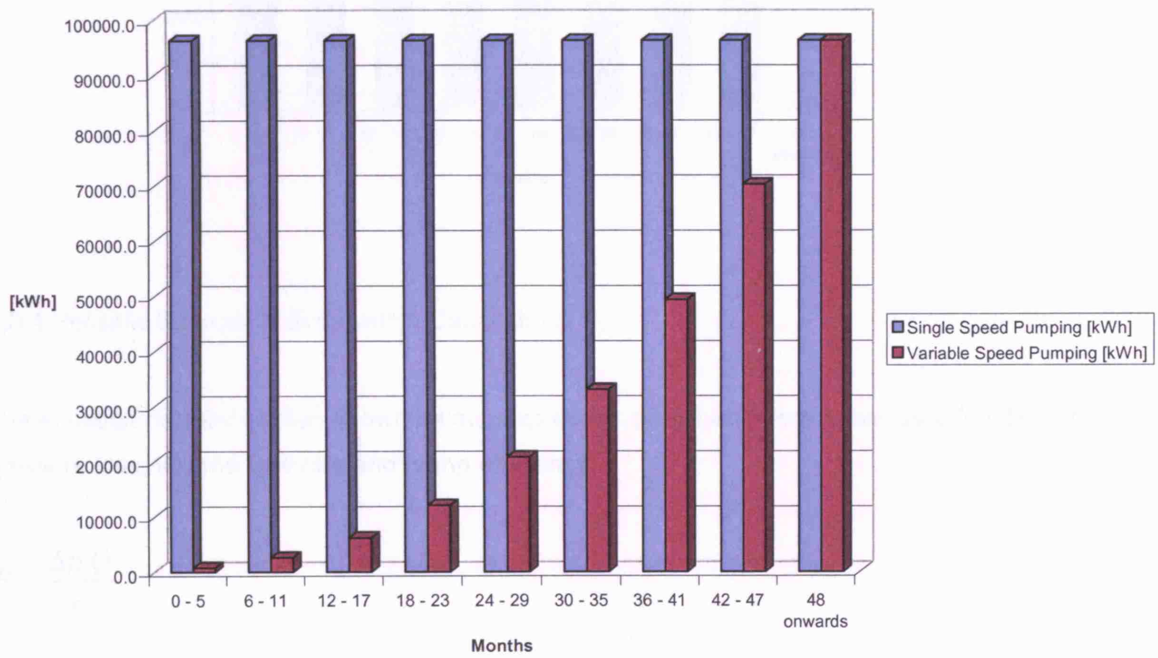
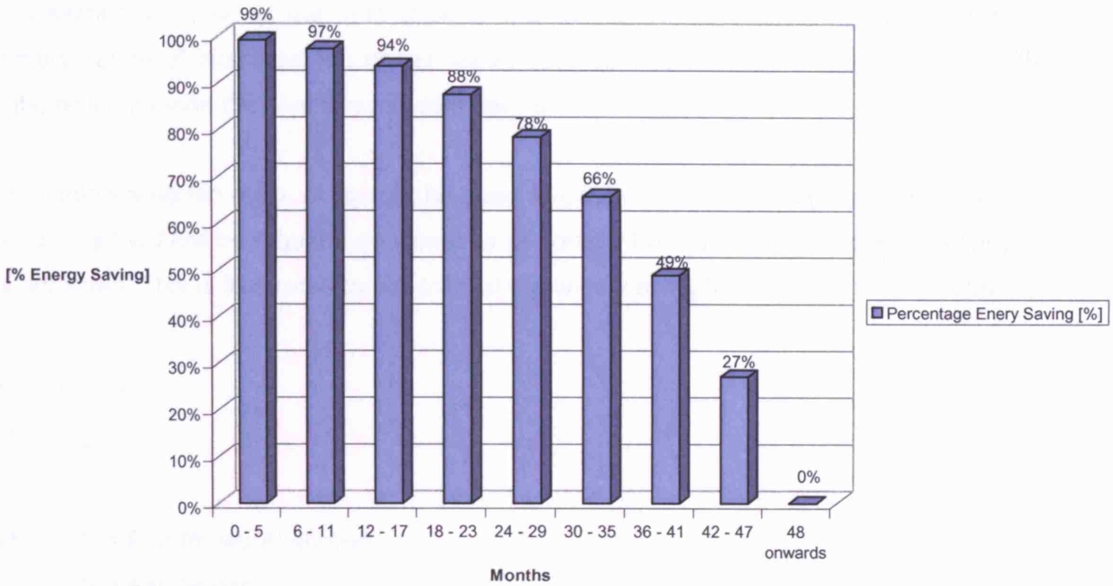


Figure 5.3.1a shown above is generated from Calculation Sheet 5.3.1a. It details the energy consumed by both the single speed pumping philosophy, as well as the variable speed during the data centre population period. It can be seen that during the early months of data centre population the energy consumed over months 0 to 5 by the single speed pump is significantly larger than the energy consumed by the variable speed alternative.

This series of time dependant energy savings is further highlighted by Figure 5.3.1b shown below.

Figure 5.3.1b - Graph Showing Energy Consumption Percentage Saving Using Variable Speed Pumping (2 Port Control) Vs Single Speed Pumping (3 Port Control) Over 0 - 48 Months



5.3.4 Possible Sources of Error within Calculations

The equation detailed earlier, shown below, expresses absorbed pump power as a function of pressure loss, volume flow rate and pump efficiency.

$$P = \frac{\Delta p.Q}{\eta}$$

Where Δp = Pressure Loss

Q = Volume Flow Rate

η = Pump Efficiency

The above calculation and its use within the spreadsheet assumes that pump efficiency remains constant. "In most situations the pump efficiency η will decrease as the pump speed drops, meaning that the true reduction in pump power will not be as great as suggested." Reference [1]. This statement indicates that some of the % energy saving figures, particularly those early in the time period (i.e. those at lower pump speeds), may be slightly inaccurate. Nonetheless, given the magnitude of the calculated energy savings it is fair to assume a sizeable energy saving would still be achieved. To eliminate any further doubt regarding the calculated energy savings, detailed dynamic analysis would be required. This is considered to be outside the scope of this report.

5.4 Variable Speed Fans within Air Handling Plant

This section will look at the fans used in the air handling plant within data centres. The primary cause of absorbed fan power within data centres relating to fans is via the CRAC units, which provide the coolth to the data centre.

A variable speed fan must, of course, be sized to deal with the full design load, however if this design load will not be required on a regular or constant basis, significant energy savings can be achieved. This is illustrated by the third of the widely accepted fan laws shown below.

$$\frac{P_2}{P_1} \propto \left(\frac{S_2}{S_1}\right)^3$$

Where P = Fan Power Absorbed
S = Fan Speed.

Once again, given that plant is normally installed to a '2N' level it is often possible to utilise all of the installed fans, as opposed to a smaller number of fans but at a reduced volume. This often provides energy savings, even given that more fans are in operation, such as is the energy saved running each fan at the reduced capacity.

5.4.1 Different Types of Fans Assessed

In addition to analysing the performance of using traditional 'run / standby' arrangements as well as 'hot standby' arrangements 2 types of fan unit will be analysed. These will be as follows:

- Centrifugal Forward Curved Impeller Fans (Inverter Belt Drive Control)
- Plug Fans / Centrifugal Backward Curved Impeller Fans (Electronically Commutated Direct Drive Control)

5.4.1.1 Centrifugal Forward Curved Impeller Fans

"The general-purpose centrifugal fan has a volute or scroll shaped casing as illustrated in the figure below. The air enters through a circular inlet adapted for use with either open inlet from the atmosphere or duct connection. The outlet is rectangular and intended for duct or diffuser connection only (without such a connection the performance would suffer). Three or four alternative impellers are often available with differing performance characteristics. The impeller may be direct driven by an external motor, but is more commonly belt driven for flexibility in speed selection.

For upwards of three-quarters of a century the (forward curved) multi-vane impeller has been used for the established general purpose type of centrifugal fan in small and medium sizes. It provides the most compact and probably the quietest form, and the most competitive in first cost. The chief disadvantage is a limited efficiency – 60 to 70% at most – and a steeply rising power characteristic towards free flow, necessitating a relatively high powered driving motor." [Reference 8]

The CRAC unit selections undertaken by Stulz Gmbh incorporating centrifugal forward curved fans are controlled by a belt drive using inverter speed control.

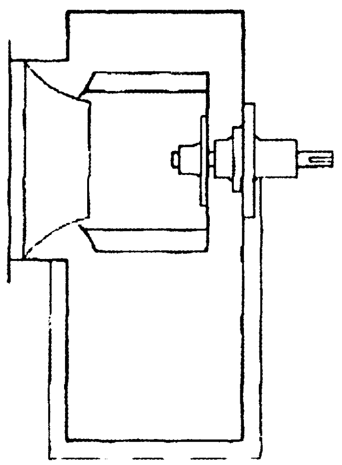


Figure 5.4.1d – Cross Section of General Purpose Centrifugal Fan

5.4.1.2 Plug Fans / Centrifugal Backward Curved Impeller Fans

"In these (fans) the outer, trailing, edges of the blades are inclined backwards rather than forwards, and the curvature is reversed with respect to the direction of rotation. Again there is little change in the relative air velocity with the impeller, but there is no reversal of direction, so that air leaves the impeller with comparatively low velocity. For a given duty the casing is bulkier and the tip speed higher than the forward curved requires. However the low velocities and, therefore friction loss, together with the avoidance of flow separation, make backward curved fans much more efficient, 80 to 85% being attainable in general purpose models.

The geometry of the blading permits fewer blade of much greater radial depth to be used, so that pressure development is largely by centrifugal action." [Reference 8].

Another advantage is their non-overloading power characteristic, by which is meant a power input which does not peak at either free flow or no flow." [Reference 8].


#

Photo 5.4.1.2a – Backward Curved / Plug Fan (Courtesy of Stulz Gmbh)

The CRAC unit selections undertaken by Stulz Gmbh incorporating centrifugal backward curved fans are controlled by a direct drive using Electronically Commutated (EC) speed control with a 0 to 10V input control.

Commutation refers to the action of production of the optimum amount of motor torque at the shaft by applying current to the relevant motor phase. This is achieved by reversing the direction of (an alternating electric current) each half-cycle to produce a unidirectional current.

In the Stulz CyberAir unit this is achieved using a brushless motor, meaning there is reduced wear and tear in addition to simplified maintenance and minimal friction losses.

Commutation is achieved by switching electronics using rotor position information, obtained by Hall sensors, optical encoder, or a resolver.

5.4.2 Fan Analysis

The fan analysis selections were undertaken with Stulz Gmbh to determine CRAC unit performance at a set series of criteria. The fan analysis had two purposes. Firstly, to determine the performance of CRAC units, at chilled water temperatures and to measure the increase in absorbed fan power, with increased chilled water temperature using backward curved centrifugal fans or plug fans.

Therefore in order to undertake this, a nominal 65 kW CRAC unit was selected as being a realistic option. This results in a requirement for 16 No. CRAC units as an 'N' requirement.

The performance of the unit was then measured via the Stulz Gmbh selection programme at 3 No. chilled water conditions. Namely:

- 6°C Flow / 12°C Return
- 8°C Flow / 14°C Return
- 10°C Flow / 16°C Return

All of the results produced by Stulz included the following for a EC / Plug fan:

- Actual Net Cooling at 100% Cooling Duty
- Air Volume Flow Rate at 100% Cooling Duty
- Run Current at 100% Cooling Duty
- Actual Net Cooling at 50% Cooling Duty
- Air Volume Flow Rate at 50% Cooling Duty
- Run Current at 50% Cooling Duty

The total absorbed fan power (K_e) at each ChW flow and return temperature range for both 100% and 50% cooling duties was then calculated using the following formula using the run currents provided by Stulz Gmbh.

$$K_e = \sqrt{ph} \times V_{line} \times I_{line} \times pf$$

Where ph = Number of phases

V_{line} = Line Voltage (volts)

I_{line} = Line Current (amperes)

pf = Power Factor (assumed to be 0.9)

The total absorbed fan power is then converted to an annual energy consumption within the authors developed spreadsheet.

The second part of the analysis was to carry out an identical series of calculations but selecting the CRAC units using forward curved centrifugal fans as opposed to backward curved. This would enable a direct comparison between the two different series of selections.

Given the notional 'N' capacity of 16 CRAC units, there will be a requirement for redundancy in the number of installed CRAC units. Given the high level of redundancy required by data centre operators, these would generally be installed on a '2N' basis. This will result in a notional installed capacity of 32 CRAC units. As with all previous calculations the results presented within this section investigate the potential energy savings achieved by using the plant in a 'hot standby' control arrangement.

Both series of selections were then analysed in relation to annual energy performance, and the related financial and carbon costs.

The results of the analysis are presented in a graphical format as shown in Figures 5.4.1a, b and c (shown below and in the appendices).

Figure 5.4.1a - Graph Showing Annual Energy Consumption for Electonically Commutated Fans (65 kW CRAC Unit) at Varying Water Temperatures in both 'N' and '2N' Running Arrangements

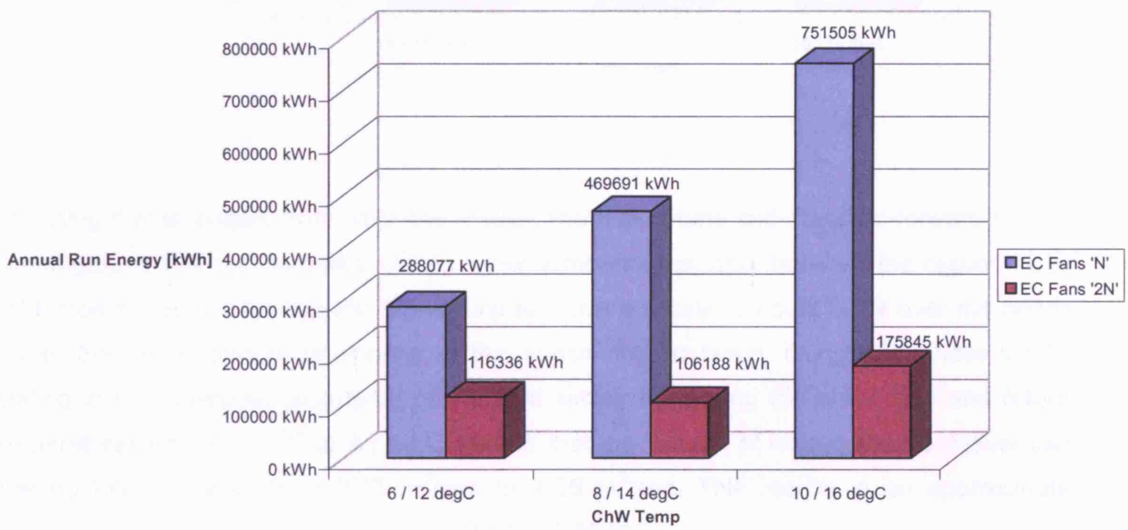
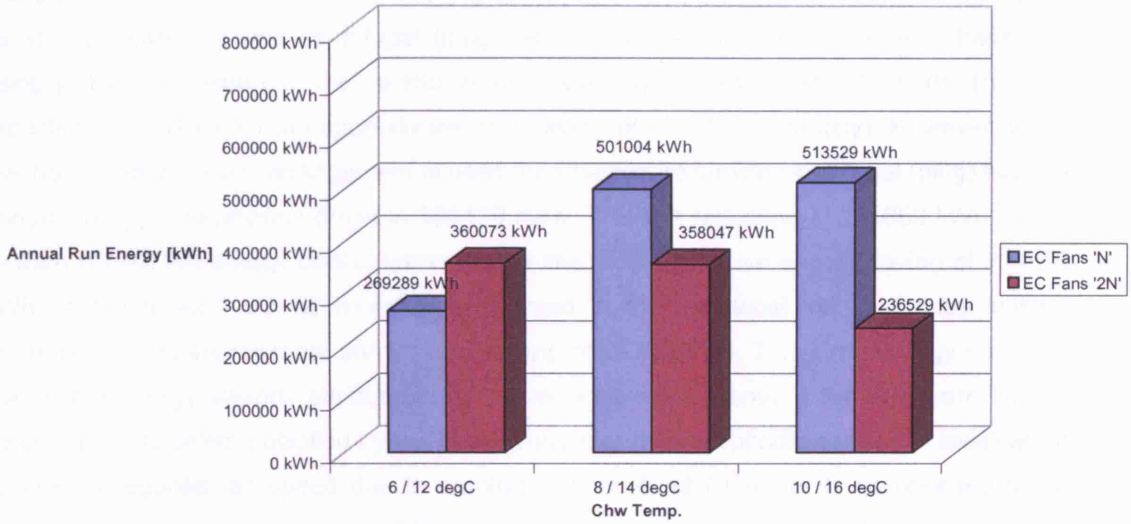


Figure 5.4.1b - Graph Showing Annual Energy Consumption for Centrifugal Fans (65 kW CRAC Unit) at Varying Water Temperatures in both 'N' and '2N' Running Arrangements



One thing that is evident from all of the results, (both plug fans and standard forward curved centrifugal), is that the increased chilled water temperatures, and therefore the higher mean coil temperatures necessitate the CRAC unit to move a greater amount of air over the coil to deliver the same amount of cooling to the space. For instance, calculation sheet 5.4.1a relating to EC controlled plug fans, shows that simply increasing the water flow and return temperatures from 6 / 12°C to 8 / 14°C means that the volume of air required to deliver just over 65 kW increases from 3.75 m^3/sec to 4.55 m^3/sec. This results in an approximate increase in absorbed fan power from 2.06 kW to 3.35 kW.

This causes a slight problem in relation to achieving both chiller free cooling and low absorbed fan power. Whilst the elevated ChW temperatures that ensure maximum 'free cooling' can be achieved at the primary refrigeration plant, this leads to an increase in absorbed fan power at the indoor terminal (CRAC) units.

Figures 5.4.1a and 5.4.1b show the two separate types of fan at the three separate water temperature ranges (6 / 12°C, 8 / 14°C and 10 / 16°C) in both a traditional 'run / standby' control arrangement and a 'hot standby' control arrangement. The purpose of these two figures is, firstly, to show the energy savings achievable by switching from forward to backward curved (plug) fans. Secondly, to demonstrate the further increase in energy saving possible by using a 'hot standby' control arrangement.

Based on a notional constant load of 1000 kW using forward curved centrifugal fans at 8 / 14°C running in a traditional 'run / standby' control arrangement uses 501004 kWh annually (see Figure 5.4.1b), This can be reduced to 358047 kWh annually simply by switching to a 'hot standby' control arrangement. For the same ChW temperature condition (8 / 14°C) but using a backward curved centrifugal (plug) fan the annual energy consumption based on using just the 'N' plant (i.e. the traditional 'run / standby' control) is 469691 kWh. This is a reduction of 31313 kWh (an approximate cost saving of £2191.91 annually). However, when the 'hot standby' control arrangement is used for a backward curved centrifugal (plug) fan, the annual energy consumption drops to 106188 kWh. This is a reduction of 251859 kWh on the forward curved fan energy consumption like for like, and a massive energy saving of 394816 kWh on the forward curved centrifugal fan used in the traditional 'run / standby' control arrangement (an approximate annual cost saving of £27637.12). This large energy saving is due to the energy savings attributable to the reduced run currents of the backward curved centrifugal fans being multiplied by the cubed nature of the multiplication (see fan laws) when running at reduced fan speed due to running '2N' levels of CRAC units as opposed to 'N' levels. These energy savings are all the more remarkable considering that the energy saving is still being delivered when double the number of fans are running.

Figure 5.4.1c - Graph Showing Tonnes of Carbon Saving / Expense Based Upon Varying Chilled Water Temperature for both EC and Centrifugal Fans

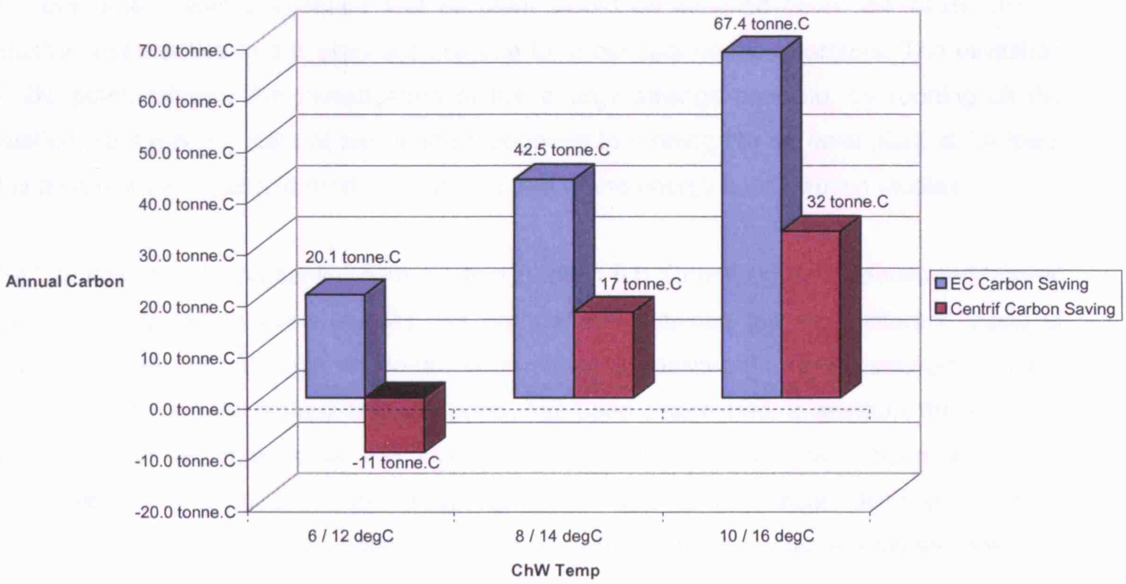


Figure 5.4.1c shows the annual carbon emission savings for both types of fans at each separate water condition when switching from a traditional 'run / standby' control arrangement to a 'hot standby' control arrangement. In the instance of the forward curved centrifugal fan 6 /12°C we can see there has been an increase of approximately 11 tonnes of carbon annually.

6.0 Conclusions and Recommendations

Data centres are becoming increasingly prevalent as our reliance on IT based technologies increases.

Given the large sums of money that these data centres either control or generate, data centre operators are more concerned with ensuring there is no down time, leading to financial loss, than they are with minimising energy consumption and reducing carbon emissions. However, as data centres require more and more cooling due to the advent of higher density blade server cooling, more data centre operators are being forced to look at how to reduce their energy load. This is, in part, forced upon them by the governments targeting of CO_2 emissions, and the necessity to 'appear green' in a marketplace becoming awakened to the impact we and the companies we use have on the environment.

To facilitate calculations a notional data centre was detailed in section 2.1 to allow like for like comparison, all results have been presented in annual energy consumption, or saving achievable (in kWh). In addition to this, where applicable, atmospheric annual carbon emissions have also been presented (in tonnes of carbon). Several assumptions were made to allow for simplicity of calculation; these included the stepped population of the data centre. The assumption was also made that all plant would be installed on a '2N' basis; this is industry best practice and is standard practice for most data centre operators. The utilisation of '2N' plant, allowed the investigation of the energy savings possible, by running all the installed plant running plant at part load as opposed to running the 'N' level plant at full load. This area of investigation formed a significant part of the energy consumption studies.

The tabulated results presented below within Section 6.0 'Conclusions & Recommendations' have been created to allow like for like comparison between the four different areas of analyses. Where possible the maximum energy saving possible (i.e. what technology within the section delivers the highest energy saving) has been determined. In addition, the left hand column of the tabulated data contains a guide to which values have been added, or subtracted, from each other by assigning each value a numerical identifier. Where geographical location has no influence on energy savings (pump and fan energy savings), the energy savings have been repeated for each location within the same table, for format consistency.

6.1 Conclusions & Recommendations - Chillers

Table 6.1a – Summary of Different Option Chiller Energy Analysis

No	Description	London		Edinburgh		Manchester	
		8 / 14°C	10 / 16°C	8 / 14°C	10 / 16°C	8 / 14°C	10 / 16°C
1	Annual Base Energy Required - 4 Standard Chillers @ 100% i.e. 'run / standby'	9408240 kWh	9408240 kWh	9408240 kWh	9408240 kWh	9408240 kWh	9408240 kWh
2	Annual Total Energy Saved - 4 Free Cooling Chillers @ 100% i.e. 'run / standby'	822246 kWh	1352378 kWh	1273915 kWh	1919139 kWh	958632 kWh	1531188 kWh
3	Annual Total Energy Saved - 8 Free Cooling Chillers @ 50% i.e. "hot standby'	1641832 kWh	2694219 kWh	2484538 kWh	3708314 kWh	1915761 kWh	3049458 kWh
1 - 2	Annual Total Energy Required - 4 Free Cooling Chillers @ 100% i.e. 'run / standby'	8585994 kWh	8055862 kWh	8134325 kWh	7489101 kWh	8449608 kWh	7877052 kWh
1 - 3	Annual Total Energy Required – 8 Free Cooling Chillers @ 50% i.e. 'hot standby'	7766408 kWh	6714021 kWh	6923702 kWh	5699926 kWh	7492479 kWh	6358782 kWh
1 - (1 - 3)	Maximum Energy Saving	1641832 kWh	2694219 kWh	2484538 kWh	3708314 kWh	1915761 kWh	3049458 kWh

All of the evidence suggests that varying energy savings are available using free cooling chillers dependant on geographical location, flow and return ChW temperature range and plant control regime. Using the 'redundant' plant to 'free cool' maximises the free cooling available, at any given temperature. Therefore, it is strongly advisable for data centre operators to employ a 'hot standby' plant arrangement, in favour of a traditional 'run / standby' plant arrangement, which would leave 50% of the plant (in the case of a '2N' plant installation) totally redundant, therefore performing no free cooling.

Of the three separate geographical locations Edinburgh provides the best opportunity for free cooling due to the increased frequency of cold temperatures. Using a 'run / standby' plant arrangement (at water flow and return temperatures of 10 / 16°C) it is possible to achieve

20% of the total cooling required on an annual basis via free cooling. This figure rises to 39% when a 'hot standby' plant arrangement is used. This increase in free cooling achievable requires no additional plant capital expenditure. The only possible marginal increase in capital expenditure could be in controls software. Most modern controls packages would be capable of dealing with this slight increase in required functionality at no additional cost.

The levels of free cooling available expressed within this section are a best case scenario of the analysis (i.e. Edinburgh with an elevated chilled water temperature). As an example London, at the same chilled water temperature flow and return range, would achieve free cooling levels as a percentage of the total required for a 'run / standby' and 'hot standby' plant arrangements of 14% and 29% respectively. This, whilst evidently not as good as Edinburgh, should be considered as a reasonable percentage of the total cooling. Therefore, the free cooling option should be considered for a new build or retrospective upgrade for all data centres within London. In all UK loads it is preferable to have a 24 hour requirement for cooling to maximise free cooling as for the majority of the time temperatures which allow free cooling occur during the evening when developments such a offices (with a '9 to 5' cooling demand) would be closed.

Utilising the aforementioned best case free cooling scenario ('hot standby' plant arrangement and elevated chilled water temperature), a date centre situated in London, Edinburgh and Manchester could reduce its carbon emissions by 315, 433 and 356 tonnes of carbon annually respectively. This commitment to not polluting the environment and would help any company portray a green image - a powerful sales tool in today's environment conscious world.

The author estimates that a free cooling chiller would cost in the region of 10 to 20% over and above the cost of a standard chiller. This cost increase percentage could amount to a significant capital financial expenditure if the chiller installation was for many megawatts, however, many megawatts of chillers installed, would of course offer the greatest free cooling potential. To determine whether a free cooling chiller should be utilised in any geographical location it would be necessary to undertake a life cycle costing to determine the financial viability. A life cycle costing does not take into account environmental impact.

The CIBSE TRY file weather data was, typically, complied between 1983 and 2004. Over recent years weather has generally become more extreme. Whilst, this leads to colder winters, it also leads to warmer summers. It would be necessary to determine weather data over recent history as well as predicting the weather in years to come to be able to accurately model the free cooling potential of a data centre with a life expectancy of approximately 15 years.

6.2 Conclusions & Recommendations – Refrigerant Cooling Systems

Table 6.1b – Summary of Different Option Trox CO_2 Energy Analysis				
No	Description	London	Edinburgh	Manchester
4	Annual Total Energy Required – 2 Trox / Star Chillers @ 100% i.e. 'run / standby'	1299368 kWh	1226097 kWh	1258405 kWh
5	Annual Total Energy Required – 4 Trox / Star Chillers @ 50% i.e. 'hot standby'	1132727 kWh	1067721 kWh	1092884 kWh
4 - 5	Maximum Energy Saving	166641 kWh	158376 kWh	165521 kWh

The Trox AITCS CO_2 refrigerant system is one of a handful of refrigerant based systems designed for the IT market.

Table 6.1b above indicates that there is only a slight effect on system efficiency due to the external ambient temperature at the outdoor central refrigeration plant; this is demonstrated by the difference between London and Edinburgh energy expenditure of 73271 kWh when two chillers are utilised in a 'run / standby' arrangement. This equates to a 13.6% energy saving in siting the data centre in Edinburgh. When four chillers are used, but at 50% load, the energy saving difference between the two same geographic locations is 65006 kWh. This is a reduced percentage energy saving of 5.7%. The reduced impact on system energy usage by low temperature frequency can be attributed to the fact that the external ambient temperature only has a bearing on the efficiency of the air cooled condenser (i.e. a reduced external ambient temperature causes an increase in condenser efficiency 'η'), located within the control refrigeration plant. The internal energy absorption is due to the latent phase change which absorbs heat from the surrounding environment as the pressure decreases, and causes the phase to change.

Once again, as with the chillers, running the plant at part load offers a significant percentage absorbed energy saving. In the case of the refrigerant system, the percentage energy saving achieved via running the plant at part load, is in the same region as than that due to geographic location. For instance in Edinburgh switching from a 'run / standby' central refrigeration plant arrangement to a 'hot standby' arrangement offer a percentage energy saving of approximately 12.9%.

Figure 5.2.1a shows that Coefficients of System Performance (COSP's) in excess of 9.0 can be achieved at part loads in the region of 40 to 50%.

An added benefit of using CO_2 as a refrigerant is that it has an ODP equal to 0, and a GWP, equal to 1. The GWP can be reduced to zero if the CO_2 is recovered from industrial processes, therefore introducing no CO_2 to the environment. Any company with aspirations to environmental credibility should ensure all CO_2 used within its processes is recovered. The usage of the CO_2 as a refrigerant in the Trox AITCS system is only within the internal data hall side of the system. The central refrigeration plant still utilises the same refrigerant, R134a, as a standard air cooled chiller. Whilst R134a has an ODP equal to zero, it has a GWP equal to 1300. This very high GWP means that any refrigerant escape due to maintenance or damage will have a negative effect on the environment and contribute to global warming.

6.3 Conclusions & Recommendations – Pumps

Table 6.1c – Summary of Different Option Pump Energy Analysis

No	Description	London		Edinburgh		Manchester	
		Primary Pump	Secondary Pump	Primary Pump	Secondary Pump	Primary Pump	Secondary Pump
6	Single Speed Pump Average Annual Energy Cost – Based on DC** Population Period	* 216777 kWh	* 216777 kWh	* 216777 kWh	* 216777 kWh	* 216777 kWh	* 216777 kWh
7	Variable Speed Pump Average Annual Energy Cost – Based on DC** Population Period	* 72837 kWh	* 72837 kWh	* 72837 kWh	* 72837 kWh	* 72837 kWh	* 72837 kWh
6 - 7	Maximum Energy Saving	143940 kWh	143940 kWh	143940 kWh	143940 kWh	143940 kWh	143940 kWh
		Primary + Secondary = 287880 kWh		Primary + Secondary = 287880 kWh		Primary + Secondary = 287880 kWh	

Notes

* Value averaged over 4 years of data population.
** Data Centre = Data Centre

Two different pumping philosophies were analysed within the report. Namely single speed pumping used in conjunction with 3-port valve control, and variable speed pumping, used in conjunction with 2-port valve control.

At first glance, the constant cooling profile / load of a data centre would lend itself to the usage of a single speed pump sized to meet the constant load. However on day one of a data centre opening the installed IT equipment will be a fraction of the data centre capacity. Therefore, the cooling requirement will be a fraction of the installed cooling capacity. During this period, when the data centre is running at less than full capacity there is a case to be made for utilising variable speed pumping.

The calculations undertaken within this report assume that the data centre will be fully populated over a period of 4 years (48 months). This assumption is based on the author's industry experience. For the purpose of the calculations the assumption is made that the data centre is 20% populated on day one, with 10% stepped increases in cooling load every 6 months.

The results presented above in Table 6.1c show the mean annual energy saving over 4 years, in reality however, the energy savings are weighted in favour of the early part of the population period. For instance, the calculations show that between months 0 to 5 inclusive, the energy saved using a variable speed pump, as opposed to using a single speed pump, could be as high as 99%. This huge energy saving is due to any drop in volume flow rate resulting in a drop in pump power absorbed, in line with an inverse cubed relationship as per the pump laws. However, even in months 42 to 57 inclusive (the final 6 month period prior to full population), a 27% energy saving is achieved. Over the course of the 4 year population period the total energy saving is assessed to be 66%.

The benefits of using variable speed pumping are negated when the data centre is fully populated, as long as the cooling load remains at its peak and constant as there are no energy savings from running the pump at part load. Given that a data centre is expected to function for in the region of 15 years, there will only be the possibility of significant energy savings for approximately 26% of the design life of the data centre.

There is an increased cost associated with the inverters, actuators, differential pressure valves, and controls required to install a variable speed pumping circuit. Once again to fully determine the financial implications of the viability of the installation it would be necessary to undertake a project specific life cycle costing. The environmental impact of the related energy and carbon savings are not generally included in life cycle costing, and therefore if the decision to install either pumping option was to be made solely on life cycle costing, it would be a measure of affordability with no weighting to environmental impact. To ensure the environmental concerns were taken into account, regardless of cost, it would be necessary to force data centre operators to meet certain environmental standards via statutory regulations.

6.4 Conclusions & Recommendations – Fans

Table 6.1d – Summary of Different Option Fan Energy Analysis

No	Description	London			Edinburgh			Manchester		
		6 /12 °C	8 /14 °C	10 /16 °C	6 /12 °C	8 /14 °C	10 /16 °C	6 /12 °C	8 /14 °C	10 /16 °C
8	Forward Curved Centrif Fan – 'N' Fans @ 100% Capacity i.e. 'run /standby'	269289 kWh	501004 kWh	513529 kWh	269289 kWh	501004 kWh	513529 kWh	269289 kWh	501004 kWh	513529 kWh
9	Forward Curved Centrif Fan – '2N' Fans @ 50% Capacity i.e. 'hot standby'	360073 kWh	358047 kWh	236529 kWh	360073 kWh	358047 kWh	236529 kWh	360073 kWh	358047 kWh	236529 kWh
10	Backward Curved Centrif Fan – 'N' Fans @ 100% Capacity i.e. 'run /standby'	288077 kWh	469691 kWh	175845 kWh	288077 kWh	106188 kWh	175845 kWh	288077 kWh	106188 kWh	175845 kWh
11	Backward Curved Centrif Fan – '2N' Fans @ 50% Capacity i.e. 'hot standby'	116336 kWh	106188 kWh	751505 kWh	116336 kWh	469691 kWh	751505 kWh	116336 kWh	469691 kWh	751505 kWh
8 - 9	Forward Curved Centrif - Maximum Energy Saving	-90784 kWh	142957 kWh	276730 kWh	-90784 kWh	142957 kWh	276730 kWh	-90784 kWh	142957 kWh	276730 kWh
10 - 11	Backward Curved Centrif - Maximum Energy Saving	171741 kWh	326734 kWh	474775 kWh	171741 kWh	326734 kWh	474775 kWh	171741 kWh	326734 kWh	474775 kWh

The fan analysis that was undertaken compared two different types of fans, the forward curved centrifugal fan and the backward curved centrifugal or plug fan. These fans have efficiencies in the range of 60 to 70%, and, 80 to 85% respectively.

These fans were analysed as part of a Stulz Gmbh CRAC unit, therefore, they were assessed at three separate water temperature ranges, namely, 6 / 12°C, 8 / 14°C and 10 / 16°C. The purpose of measuring fan performance at differing water temperatures was to determine to what degree the increase in mean coil temperature, affected the volume of air that needed to be passed across the coil, to deliver the same amount of net sensible cooling. For example, the higher the mean coil temperature, the greater the volume of air that needs to pass over that coil to deliver the same quantity of cooling to the space. As the volume flow rate increases, so the fan speed must increase. This increase in fan speed results in a greater absorption of power by the CRAC unit fan. To maximise CRAC unit fan efficiency, it is required that the fan operates at a low fan speed. There is a minimum fan speed in CRAC units, as a certain floor pressure must be maintained to enable to the air to leave the floor outlet at a velocity sufficient to ensure cooling reaches the top of the IT cabinet. Also, DX CRAC units have minimum fan speed requirements to ensure the refrigerant coil does not fracture due to over cooling.

In addition, to allow comparison between running the fans at part and full load (i.e. 'run / standby' or 'hot standby' control), all of information was requested from Stulz at 100% of CRAC unit cooling capacity, and also at 50%.

Table 6.1d shows that with one exception (out of a total of six possible results), running the fans with a 'hot standby' control arrangement leads to an energy saving. The energy savings from running the 'hot standby' control arrangement, as opposed to the traditional 'run / standby' control arrangement, are also higher when the more efficient backward curved fans are used. This is in addition to the energy required to run backward curved fans being lower than for forward curved fans. Therefore, the most energy efficient of CRAC unit fan installation and control is to combine backward curved or plug fans, with a part load operation ('hot standby').

One result (DIS/005, 6 / 12°C a forward curved centrifugal fan), does not conform to the expected results of showing an energy saving when using a 'hot standby' control arrangement. This is due to the 50% cooling duty figure provided by Stulz being only 0.9 kW less than the 100% cooling duty figure (64.4 kW and 65.3 kW respectively). This is due to the minimum fan turn down within the CRAC unit. To alleviate this problem it would prudent to install two smaller units of equal combined duty. This was not possible if calculation uniformity was to be maintained. Further analysis and detailed design would be expected to show that backward curved centrifugal fans, in conjunction with a 'hot standby' control arrangement, offers the most energy efficient combination in all cases.

To maximise the fan efficiency (i.e. minimal volume flow rate) it is necessary to use a low water temperature range. Doing this however, in conjunction with free cooling chillers would limit the opportunity for free cooling at the chiller. If both technologies were to be used in tandem, the energy saved from the central refrigeration side of the system would run the risk of being squandered on the internal CRAC unit fan energy consumption.

To decide which technology should be used would require job specific analysis to determine which offered the greatest energy savings. It would then be wise to utilise which ever technology offered the greatest energy savings.

The installation of variable speed drives is a minor financial implication due to their inexpensive nature. The author estimates this to be in the region of £400 to £600 per CRAC unit.

7.0 References

Reference [1] Variable Speed Pumping in Heating and Cooling Circuits: Application Guide AG 14/99
The Building Services Research and Information Association (BSRIA): 1999
C J Parsoe

Reference [2] http://www.cibse.org/index.cfm?go=publications.view&PubID=332&L1=164
The Chartered Institute of Building Services Engineers
Accessed: August 2006

Reference [3] Heating & Air Conditioning of Buildings. Ninth Edition
Faber & Kell's: 2002
D R Oughton & S Hodkinson

Reference [4] http://www.cisco.com/
Accessed: August 2006

Reference [5] http://www.ace.mmu.ac.uk/eae/Global_Warming/Older/GWPs.html
Manchester Metropolitan University
Accessed July 2006

Reference [6] CO2 Mission Critical Cooling Paper, Presented to the Institute of Refrigeration Annual Conference London 2005.
Trox Advanced IT Cooling Systems (Trox AITCS): 2005
Guy Hutchins

Reference [7] http://www.co2clean.com/snowform.htm
Applied Surface Technologies
Accessed: August 2006

Reference [8] Woods Guide to Practical Fan Engineering: Sixth Impression
Flakt Woods Limited: 1992
B B Daly

Reference [9] http://www.shecco.com/artikler/Greenhouse_gas_effects.htm
Shecco Technology
Accessed: September 2006

Appendices – Raw Data & Calculations

See next page

Calculation 5.1.1a - Free Cooling Analysis Raw Data & Calculations: ChW Flow / Return Temp = 8 / 14 degC

Temp. Range	Number of hours - London (Hebtry.wfl)	Number of hours - Edinburgh (Tumtry.wfl)	Number of hours - Manchester (Riatry.wfl)	Max Chiller Load	Empirical Temp Dependant Load	% of Total Chiller Load	'N' Load - 4 No. Chillers.	'N' Load - 4 No. Chillers Free Cooling Achieved	4 Chiller Total Free Cooling - London	4 Chiller Total Free Cooling - Edinburgh	4 Chiller Total Free Cooling - Manchester	'2N' Load - 8 No. Chillers Potential Free Cooling	'2N' Load - 8 No. Chillers Free Cooling Achieved	8 Chiller Total Free Cooling - London	8 Chiller Total Free Cooling - Edinburgh	8 Chiller Total Free Cooling - Manchester
degC	Hours	Hours	Hours	kW	kW	%	kW	kW	kWh	kWh	kWh	kW	kW	kWh	kWh	kWh
-99 to -4	4	115	0	248.0	183.6	74%	992	734	2938	84456	0	1469	992	3968	114080	0
-4 to -3	4	45	8	248.0	147.5	59%	992	590	2360	26550	4720	1180	992	3968	44640	7936
-3 to -2	26	83	43	248.0	111.4	45%	992	446	11586	36985	19161	891	891	23171	73970	38322
-2 to -1	40	111	88	248.0	109.0	44%	992	436	17440	48396	38368	872	872	34880	96792	76736
-1 to 0	82	154	88	248.0	107.0	43%	992	428	35096	65912	37664	856	856	70192	131824	75328
0 to 1	91	217	140	248.0	103.9	42%	992	416	37820	90185	58184	831	831	75639	180370	116368
1 to 2	226	352	195	248.0	100.0	40%	992	400	90400	140800	78000	800	800	180800	281600	156000
2 to 3	270	439	272	248.0	95.4	38%	992	382	103032	167522	103795	763	763	206064	335045	207590
3 to 4	358	459	374	248.0	90.4	36%	992	362	129453	165974	135238	723	723	258906	331949	270477
4 to 5	365	410	452	248.0	84.5	34%	992	338	123370	138580	152776	676	676	246740	277160	305552
5 to 6	430	486	535	248.0	78.1	31%	992	312	134332	151826	167134	625	625	268664	303653	334268
6 to 7	470	548	572	248.0	71.5	29%	992	286	134420	156728	163592	572	572	268840	313456	327184
7 to 8	496	606	550	248.0	0.0	0%	992	0	0	0	0	0	0	0	0	0
8 to 9	641	579	621	248.0	0.0	0%	992	0	0	0	0	0	0	0	0	0
9 to 10	549	614	576													
10 to 11	542	497	575						822246	1273915	958632			1641832	2484538	1915761
11 to 12	523	515	592													
12 to 13	452	566	535						9408240	9408240	9408240			9408240	9408240	9408240
13 to 14	494	504	529													
14 to 15	429	410	515						London	Edinburgh	Manchester			London	Edinburgh	Manchester
15 to 16	390	354	441													
16 to 17	437	278	336						9%	14%	10%			17%	26%	20%
17 to 18	334	166	280													
18 to 19	267	106	123			Carbon Emmission Index (kg.C/kWh)			0.117	0.117	0.117			0.117	0.117	0.117
19 to 20	209	74	98													
20 to 21	153	36	60			Annual Carbon Saving (kg.C)			96203 kg.C	149048 kg.C	112160 kg.C			192094 kg.C	290691 kg.C	224144 kg.C
21 to 22	122	24	63													
22 to 23	81	5	44			Annual Carbon Saving (tonne.C)			96.2 tonne.C	149.0 tonne.C	112.2 tonne.C			192.1 tonne.C	290.7 tonne.C	224.1 tonne.C
23 to 24	80	3	16													
24 to 25	68	0	14													
25 to 26	49	1	11													
26 to 27	33	3	13													
27 to 28	21	0	1													
28 to 29	13	0	0													
29 to 30	10	0	0													
30 to 31	1	0	0													
31 to 32	0	0	0													
32 to 33	0	0	0													
33 to 34	0	0	0													
34 to 35	0	0	0													
35 to 36	0	0	0													
36 to 37	0	0	0													
37 to 38	0	0	0													
38 to 39	0	0	0													
39 to 40	0	0	0													
40 to 41	0	0	0													
41 to 42	0	0	0													
42 to 43	0	0	0													
43 to 44	0	0	0													
44 to 99	0	0	0													
	8760	8760	8760													

Calculation 5.1.2a - Free Cooling Analysis Raw Data & Calculations: ChW Flow / Return Temp = 10 / 16 degC

Temp. Range	Number of hours - London (Hebtry.wfl)	Number of hours - Edinburgh (Tumtry.wfl)	Number of hours - Manchester (Ristry.wfl)	Max Chiller Load	Empirical Temp Dependant Load	% of Total Chiller Load	'N' Load - 4 No. Chillers.	'N' Load - 4 No. Chillers Free Cooling Achieved	4 Chiller Total Free Cooling - London	4 Chiller Total Free Cooling - Edinburgh	4 Chiller Total Free Cooling - Manchester	'2N' Load - 8 No. Chillers Potential Free Cooling	'2N' Load - 8 No. Chillers Free Cooling Achieved	8 Chiller Total Free Cooling - London	8 Chiller Total Free Cooling - Edinburgh	8 Chiller Total Free Cooling - Manchester
degC	Hours	Hours	Hours	kW	kW	%	kW	kW	kWh	kWh	kWh	kW	kW	kWh	kWh	kWh
-99 to -4	4	115	0	268.4	232.4	87%	1074	930	3718	106904	0	1859	1074	4294	123464	0
-4 to -3	4	45	8	268.4	196.3	73%	1074	785	3141	35334	6282	1570	1074	4294	48312	8589
-3 to -2	26	83	43	268.4	160.2	60%	1074	641	16661	53186	27554	1282	1074	27914	89109	46165
-2 to -1	40	111	88	268.4	124.1	46%	1074	496	19856	55100	43683	993	993	39712	110201	87366
-1 to 0	82	154	88	268.4	122.2	46%	1074	489	40082	75275	43014	978	978	80163	150550	86029
0 to 1	91	217	140	268.4	119.7	45%	1074	479	43571	103900	67032	958	958	87142	207799	134064
1 to 2	226	352	195	268.4	116.3	43%	1074	465	105135	163750	90714	930	930	210270	327501	181428
2 to 3	270	439	272	268.4	112.3	42%	1074	449	121284	197199	122182	898	898	242568	394398	244365
3 to 4	358	459	374	268.4	107.7	40%	1074	431	154226	197737	161119	862	862	308453	395474	322238
4 to 5	365	410	452	268.4	102.3	38%	1074	409	149358	167772	184958	818	818	298716	335544	369917
5 to 6	430	486	535	268.4	96.6	36%	1074	386	166152	187790	206724	773	773	332304	375581	413448
6 to 7	470	548	572	268.4	90.2	34%	1074	361	169576	197718	206378	722	722	339152	395437	412755
7 to 8	496	606	550	268.4	83.3	31%	1074	333	165267	201919	183260	666	666	330534	403838	366520
8 to 9	641	579	621	268.4	75.8	28%	1074	303	194351	175553	188287	606	606	388702	351106	378574
9 to 10	549	614	576													
10 to 11	542	497	575						**1352378**	**1919139**	**1531188**			**2694219**	**3708314**	**3049458**
11 to 12	523	515	592													
12 to 13	452	566	535						9408240	9408240	9408240			9408240	9408240	9408240
13 to 14	494	504	529													
14 to 15	429	410	515						London	Edinburgh	Manchester			London	Edinburgh	Manchester
15 to 16	390	354	441													
16 to 17	437	278	336						**14%**	**20%**	**16%**			**29%**	**39%**	**32%**
17 to 18	334	166	280													
18 to 19	267	106	123			Carbon Emmission Index (kg.C/kWh)			0.117	0.117	0.117			0.117	0.117	0.117
19 to 20	209	74	98													
20 to 21	153	36	60			Annual Carbon Saving (kg.C)			**158228 kg.C**	**224539 kg.C**	**179149 kg.C**			**315224 kg.C**	**433873 kg.C**	**356787 kg.C**
21 to 22	122	24	63													
22 to 23	81	5	44			Annual Carbon Saving (tonne.C)			**158.2 tonne.C**	**224.5 tonne.C**	**179.1 tonne.C**			**315.2 tonne.C**	**433.9 tonne.C**	**356.8 tonne.C**
23 to 24	80	3	16													
24 to 25	68	0	14													
25 to 26	49	1	11													
26 to 27	33	3	13													
27 to 28	21	0	1													
28 to 29	13	0	0													
29 to 30	10	0	0													
30 to 31	1	0	0													
31 to 32	0	0	0													
32 to 33	0	0	0													
33 to 34	0	0	0													
34 to 35	0	0	0													
35 to 36	0	0	0													
36 to 37	0	0	0													
37 to 38	0	0	0													
38 to 39	0	0	0													
39 to 40	0	0	0													
40 to 41	0	0	0													
41 to 42	0	0	0													
42 to 43	0	0	0													
43 to 44	0	0	0													
44 to 99	0	0	0													
	8760	**8760**	**8760**													

Calculation Sheet 5.2.1a - Trox AITCS / Star Refrigeration Data Showing COSP for 500 kW CPA500 Unit

	Load	External Ambient Temp							
	%	35 degC	30 degC	25 degC	20 degC	15 degC	10 degC	5 degC	0 degC
COSP	100%	3.18	3.67	4.30	5.01	5.83	6.48	7.91	7.91
	90%	3.33	3.89	4.53	5.27	6.14	6.81	8.02	8.58
	80%	3.37	3.95	4.62	5.42	6.32	7.05	8.13	8.78
	70%	3.30	3.94	4.68	5.53	6.52	7.43	8.25	9.01
	60%	3.13	3.85	4.64	5.55	6.70	7.70	8.96	8.96
	50%	2.96	3.69	4.52	5.54	6.74	7.83	8.83	8.83
	40%	3.03	3.66	4.59	5.61	6.88	8.76	8.76	8.90
	30%	3.61	4.37	5.20	6.16	7.26	8.36	8.52	8.52
	20%	3.06	3.79	4.52	5.34	6.41	8.18	8.18	8.18
	10%	1.00	1.20	1.50	3.70	5.70	5.68	5.68	5.68

Calculation 5.2.1b - Trox CO2 System Analysis for London (Hebtry.wfl) site for 'N' and '2N'

Temp. Range	Number of hours - London (Hebtry.wfl)	Sum Hours	100% COSP	50% COSP	Power Input - 'N' System per System	Power Input - '2N' System per System	Total Energy Consumption - 'N' System	Total Energy Consumption - '2N' System
degC	Hours	Hours	COSP	COSP	kW	kW	kWh	kWh
-99 to -4	4							
-4 to -3	4							
-3 to -2	26	156	7.91	8.83	126.4 kW	56.6 kW	19711 kWh	17668 kWh
-2 to -1	40							
-1 to 0	82							
0 to 1	91							
1 to 2	226							
2 to 3	270	1310	7.91	8.83	126.4 kW	56.6 kW	165521 kWh	148368 kWh
3 to 4	358							
4 to 5	365							
5 to 6	430							
6 to 7	470							
7 to 8	496	2586	7.91	8.83	126.4 kW	56.6 kW	326746 kWh	292886 kWh
8 to 9	641							
9 to 10	549							
10 to 11	542							
11 to 12	523							
12 to 13	452	2440	6.48	7.83	154.3 kW	63.9 kW	376547 kWh	311778 kWh
13 to 14	494							
14 to 15	429							
15 to 16	390							
16 to 17	437							
17 to 18	334	1637	5.83	6.74	171.5 kW	74.2 kW	280750 kWh	242984 kWh
18 to 19	267							
19 to 20	209							
20 to 21	153							
21 to 22	122							
22 to 23	81	504	5.01	5.54	199.4 kW	90.2 kW	100515 kWh	90907 kWh
23 to 24	80							
24 to 25	68							
25 to 26	49							
26 to 27	33							
27 to 28	21	126	4.30	4.52	232.6 kW	110.6 kW	29306 kWh	27865 kWh
28 to 29	13							
29 to 30	10							
30 to 31	1							
31 to 32	0							
32 to 33	0	1	3.67	3.69	272.6 kW	135.3 kW	273 kWh	271 kWh
33 to 34	0							
34 to 35	0							
35 to 36	0							
36 to 37	0							
37 to 38	0	0	3.18	2.96	314.4 kW	169.1 kW	0 kWh	0 kWh
38 to 39	0							
39 to 40	0							
40 to 41	0							
41 to 42	0	8760					1299368 kWh	1132727 kWh
42 to 43	0							
43 to 44	0						152026 kg.C	132529 kg.C
44 to 99	0							
	8760						152 tonne.C	133 tonne.C

Input Information	
'N' System Requirement	1000 kW
'2N' System Requirement	500 kW
Carbom Emission Index (kg C/kWh) #	0.117

Calculation 5.2.1c - Trox CO2 System Analysis for Edinburgh (Turnhouse.wfl) site for 'N' and '2N'

Temp. Range	Number of hours - Edinburgh (Turnhouse.wfl)	Sum Hours	100% COSP	50% COSP	Power Input - 'N' System per System	Power Input - '2N' System per System	Total Energy Consumption - 'N' System	Total Energy Consumption - '2N' System
degC	Hours	Hours	COSP	COSP	kW	kW	kWh	kWh
-99 to -4	115							
-4 to -3	45							
-3 to -2	83	508	7.91	8.83	126.4 kW	56.6 kW	64187 kWh	57535 kWh
-2 to -1	111							
-1 to 0	154							
0 to 1	217							
1 to 2	352							
2 to 3	439	1877	7.91	8.83	126.4 kW	56.6 kW	237162 kWh	212586 kWh
3 to 4	459							
4 to 5	410							
5 to 6	486							
6 to 7	548							
7 to 8	606	2833	7.91	8.83	126.4 kW	56.6 kW	357955 kWh	320861 kWh
8 to 9	579							
9 to 10	614							
10 to 11	497							
11 to 12	515							
12 to 13	566	2492	6.48	7.83	154.3 kW	63.9 kW	384572 kWh	318422 kWh
13 to 14	504							
14 to 15	410							
15 to 16	354							
16 to 17	278							
17 to 18	166	978	5.83	6.74	171.5 kW	74.2 kW	167730 kWh	145167 kWh
18 to 19	106							
19 to 20	74							
20 to 21	36							
21 to 22	24							
22 to 23	5	68	5.01	5.54	199.4 kW	90.2 kW	13562 kWh	12265 kWh
23 to 24	3							
24 to 25	0							
25 to 26	1							
26 to 27	3							
27 to 28	0	4	4.30	4.52	232.6 kW	110.6 kW	930 kWh	885 kWh
28 to 29	0							
29 to 30	0							
30 to 31	0							
31 to 32	0							
32 to 33	0	0	3.67	3.69	272.6 kW	135.3 kW	0 kWh	0 kWh
33 to 34	0							
34 to 35	0							
35 to 36	0							
36 to 37	0							
37 to 38	0	0	3.18	2.96	314.4 kW	169.1 kW	0 kWh	0 kWh
38 to 39	0							
39 to 40	0							
40 to 41	0							
41 to 42	0	**8760**						
42 to 43	0							
43 to 44	0							
44 to 99	0							
	8760							

1226097 kWh	1067721 kWh
143453 kg.C	124923 kg.C
143 tonne.C	125 tonne.C

Input Information	
'N' System Requirement	1000 kW
'2N' System Requirement	500 kW
Carbom Emission Index (kg.C/kWh) #	0.117

Calculation 5.2.1d - Trox CO2 System Analysis for Manchester (Riatry.wfl) site for 'N' and '2N'

Temp. Range	Number of hours - Manchester (Riatry.wfl)	Sum Hours	100% COSP	50% COSP	Power Input - 'N' System per System	Power Input - '2N' System per System	Total Energy Consumption - 'N' System	Total Energy Consumption - '2N' System
degC	Hours	Hours	COSP	COSP	kW	kW	kWh	kWh
-99 to -4	0							
-4 to -3	8							
-3 to -2	43	227	7.91	8.83	126.4 kW	56.6 kW	28682 kWh	25710 kWh
-2 to -1	88							
-1 to 0	88							
0 to 1	140							
1 to 2	195							
2 to 3	272	1433	7.91	8.83	126.4 kW	56.6 kW	181062 kWh	162299 kWh
3 to 4	374							
4 to 5	452							
5 to 6	535							
6 to 7	572							
7 to 8	550	2854	7.91	8.83	126.4 kW	56.6 kW	360608 kWh	323239 kWh
8 to 9	621							
9 to 10	576							
10 to 11	575							
11 to 12	592							
12 to 13	535	2746	6.48	7.83	154.3 kW	63.9 kW	423769 kWh	350878 kWh
13 to 14	529							
14 to 15	515							
15 to 16	441							
16 to 17	336							
17 to 18	280	1278	5.83	6.74	171.5 kW	74.2 kW	219181 kWh	189697 kWh
18 to 19	123							
19 to 20	98							
20 to 21	60							
21 to 22	63							
22 to 23	44	197	5.01	5.54	199.4 kW	90.2 kW	39289 kWh	35533 kWh
23 to 24	16							
24 to 25	14							
25 to 26	11							
26 to 27	13							
27 to 28	1	25	4.30	4.52	232.6 kW	110.6 kW	5815 kWh	5529 kWh
28 to 29	0							
29 to 30	0							
30 to 31	0							
31 to 32	0							
32 to 33	0	0	3.67	3.69	272.6 kW	135.3 kW	0 kWh	0 kWh
33 to 34	0							
34 to 35	0							
35 to 36	0							
36 to 37	0							
37 to 38	0	0	3.18	2.96	314.4 kW	169.1 kW	0 kWh	0 kWh
38 to 39	0							
39 to 40	0							
40 to 41	0							
41 to 42	0	8760					1258405 kWh	1092884 kWh
42 to 43	0							
43 to 44	0						147233 kg.C	127867 kg.C
44 to 99	0							
	8760						147 tonne.C	128 tonne.C

Input Information	
'N' System Requirement	1000 kW
'2N' System Requirement	500 kW
Carbom Emission Index (kg C/kWh)	0.117

Calculation 5.3.1a - Chilled Water Pumping Energy Consumed Calculation - Single Speed Pumping

Based on Theoretical Stepped Population of Data Centre - See Section 2.1

Months	Data Centre IT Populated	Data Centre Cooling Load	Chilled Water Flow Rate Required	kilowatt Rating of Pump	Total Energy Consumed within Stepped Period
	%	kW	l/sec	kW	kWh
0 - 5	100%	1000	39.68	22.00	96345.3
6 - 11	100%	1000	39.68	22.00	96345.3
12 - 17	100%	1000	39.68	22.00	96345.3
18 - 23	100%	1000	39.68	22.00	96345.3
24 - 29	100%	1000	39.68	22.00	96345.3
30 - 35	100%	1000	39.68	22.00	96345.3
36 - 41	100%	1000	39.68	22.00	96345.3
42 - 47	100%	1000	39.68	22.00	96345.3
48 onwards	100%	1000	39.68	22.00	96345.3

Technical Information	
Max Data Centre Cooling Load	1000 kW
Running Current of Pump at Full Capacity	49.23 A
Supply Voltage	415 V
No. of Phases	3
Power Factor	0.90
Max Power Absorbed by Pump (100% Load)	22.00 kW
Density of Water	1000 kg/m³
Specific Heat Capacity of Water	4.2 kJ/kg.k
Temperature Difference	6.0 k

Time Related Information	
No. of Hours in Calendar Year	8760 Hrs
Approx No. of Hours in Calendar Month	730 Hrs

Notes
1) Assumes Single Speed Pumping in Conjunction with 3-port control (mixing) valve arrangement.

Calculation 5.3.1b - Chilled Water Pumping Energy Consumed Calculation - Variable Speed Pumping

Based on Theoretical Stepped Population of Data Centre - See Section 2.1

Months	Data Centre IT Populated	Data Centre Cooling Load	Chilled Water Flow Rate Required	kilowatt Rating of Pump	Total Energy Consumed within Stepped Period
	%	kW	l/sec	kW	kWh
0 - 5	20%	200	7.94	0.18	770.8
6 - 11	30%	300	11.90	0.59	2601.3
12 - 17	40%	400	15.87	1.41	6166.1
18 - 23	50%	500	19.84	2.75	12043.2
24 - 29	60%	600	23.81	4.75	20810.6
30 - 35	70%	700	27.78	7.54	33046.4
36 - 41	80%	800	31.75	11.26	49328.8
42 - 47	90%	900	35.71	16.04	70235.7
48 onwards	100%	1000	39.68	22.00	96345.3

Technical Information	
Max Data Centre Cooling Load	1000 kW
Running Current of Pump at Full Capacity	49.23 A
Supply Voltage	415 V
No. of Phases	3
Power Factor	0.90
Max Power Absorbed by Pump (100% Load)	22.00 kW
Density of Water	1000 kg/m³
Specific Heat Capacity of Water	4.2 kJ/kg.k
Temperature Difference	6.0 k

Time Related Information	
No. of Hours in Calendar Year	8760 Hrs
Approx No. of Hours in Calendar Month	730 Hrs

Notes

1) Assumes Variable Speed Pumping in Conjunction with 2-port control valve arrangement.

Calculation 5.3.1c - Chilled Water Pumping Energy Consumed Calculation Comparison - Comparison Sheet

Based on Theoretical Stepped Population of Data Centre - See Section 2.1

Months	Data Centre IT Populated	Single Speed Pumping Energy Consumed	Variable Speed Pumping Energy Consumed	Energy Consumed Percentage Reduction - See Note 1
	%	kWh	kWh	%
0 - 5	20%	96345	771	99%
6 - 11	30%	96345	2601	97%
12 - 17	40%	96345	6166	94%
18 - 23	50%	96345	12043	88%
24 - 29	60%	96345	20811	78%
30 - 35	70%	96345	33046	66%
36 - 41	80%	96345	49329	49%
42 - 47	90%	96345	70236	27%
48 onwards	100%	96345	96345	0%
		867108	291348	66%
	CO2 Output	144.03 tonnes	48.39 tonnes	

	Primary Pump	Secondary Pump	Total
Total Energy Saved Using VSD =	575760 kWh	575760 kWh	1151519 kWh
Total Cost Saved Using VSD=	£ 40303.18	£ 40303.18	£ 80606.35
Carbon Emission Indices (kg.C/kWh) =	0.117	0.117	0.117
Total Carbon Emission Saving (kg.C) =	67364 kg.C	67364 kg.C	134728 kg.C
Total Carbon Emission Saving (tonne.C) =	67.36 tonnes	67.36 tonnes	134.73 tonnes

Notes
1) The energy consumption reduction percenrage figure is for a given time period i.e. 0 - 5 months.
2) Based on Electricity Cost of £0.07 per kWh
3) Based on Carbon outputof 0.117 kg.C / kWh

Calculation Sheet 5.4.1a - CRAC Unit Calculation with Electronically Commutated (EC) fans

Electronically Commutated Fans

CRAC Unit Ref.	Chw Flow + Return Temp	Air Supply + Return Temp	Required Net Cooling at 100% Cooling Duty	Actual Net Cooling at 100% Cooling Duty	Air Volume Flow Rate at 100% Cooling Duty	Run Current at 100% Cooling Duty	Power Absorbed at 100% Cooling Duty - 1 No. CRAC Unit	Actual Net Cooling at 50% Cooling Duty	Air Volume Flow Rate at 50% Cooling Duty	Run Current at 50% Cooling Duty	Power Absorbed at 50% Cooling Duty - 1 No. CRAC Unit	Annual Energy Consumption - 'N'	Annual Energy Consumption - '2N'	Difference in Annual Energy Consumption
DIS/001	6 - 12 degC	15 - 25 degC	65.0 kW	65.3 kW	3.75 m3/sec	4.6 A	2.06 kW	37.9 kW	2.2 m3/sec	0.9 A	0.42 kW	288077 kWh	116336 kWh	171741 kWh
DIS/002	8 - 14 degC	15 - 25 degC	65.0 kW	65.1 kW	4.55 m3/sec	7.5 A	3.35 kW	33.5 kW	2.2 m3/sec	0.8 A	0.38 kW	469691 kWh	106188 kWh	363503 kWh
DIS/003	10 - 16 degC	15 - 25 degC	65.0 kW	62.3 kW	5.50 m3/sec	12.0 A	5.36 kW	35.4 kW	2.7 m3/sec	1.4 A	0.63 kW	751505 kWh	175845 kWh	575660 kWh

Technical Information	
Supply Voltage	415 V
No. of Phases	3
Power Factor	0.90

Technical Information	
'N' No. of CRAC Units	16
'2N' No. of CRAC Units	32

Calculation Sheet 5.4.1b - CRAC Unit Calculation with Centrifugal fans

Centrifugal Fans

CRAC Unit Ref.	ChW Flow + Return Temp	Air Supply + Return Temp	Required Net Cooling at 100% Cooling Duty	Actual Net Cooling at 100% Cooling Duty	Air Volume Flow Rate at 100% Cooling Duty	Run Current at 100% Cooling Duty	Power Absorbed at 100% Cooling Duty - 1 No. CRAC Unit	Actual Net Cooling at 50% Cooling Duty	Air Volume Flow Rate at 50% Cooling Duty	Run Current at 50% Cooling Duty	Power Absorbed at 50% Cooling Duty - 1 No. CRAC Unit	Annual Energy Consumption - 'N'	Annual Energy Consumption - '2N'	Difference in Annual Energy Consumption
DIS/005	6 - 12 degC	15 - 25 degC	65.0 kW	65.3 kW	4.2 m3/sec	4.3 A	1.92 kW	64.4 kW	3.69 m3/sec	2.9 A	1.28 kW	269289 kWh	360073 kWh	-90784 kWh
DIS/006	8 - 14 degC	15 - 25 degC	65.0 kW	65.2 kW	5.2 m3/sec	8.0 A	3.57 kW	55.3 kW	3.69 m3/sec	2.9 A	1.28 kW	501004 kWh	358047 kWh	142957 kWh
DIS/007	10 - 16 degC	15 - 25 degC	65.0 kW	65.0 kW	6.0 m3/sec	8.2 A	3.66 kW	45.7 kW	3.69 m3/sec	1.9 A	0.84 kW	513529 kWh	236529 kWh	277000 kWh

Technical Information	
Supply Voltage	415 V
No. of Phases	3
Power Factor	0.90

Technical Information	
'N' No. of CRAC Units	16
'2N' No. of CRAC Units	32

Calculation Sheet 5.4.1c - CRAC Unit Fan Calculation Comparison Sheet

Centrifugal Fans

EC Fan CRAC References	Chilled Water Temperature	EC Fan Annual Energy Consumption 'N'	EC Fan Annual Energy Consumption '2N'	EC Fan Difference in Annual Energy Consumption	EC Fan Annual Carbon Saving / Expense (kg)	EC Fan Annual Carbon Saving / Expense (tonne)
DIS/001	6 / 12 degC	288077 kWh	116336 kWh	171741 kWh	20094 kg.C	20.1 tonne.C
DIS/002	8 / 14 degC	469691 kWh	106188 kWh	363503 kWh	42530 kg.C	42.5 tonne.C
DIS/003	10 / 16 degC	751505 kWh	175845 kWh	575660 kWh	67352 kg.C	67.4 tonne.C

Centrifugal Fan CRAC References	Chilled Water Temperature	Centrifugal Fan Annual Energy Consumption 'N'	Centrifugal Fan Annual Energy Consumption '2N'	Centrifugal Fan Difference in Annual Energy Consumption	Centrifugal Fan Annual Carbon Saving / Expense (kg)	Centrifugal Fan Annual Carbon Saving / Expense (tonne)
DIS/005	6 / 12 degC	269289 kWh	360073 kWh	-90784 kWh	-10622 kg.C	-11 tonne.C
DIS/006	8 / 14 degC	501004 kWh	358047 kWh	142957 kWh	16726 kg.C	17 tonne.C
DIS/007	10 / 16 degC	513529 kWh	236529 kWh	277000 kWh	32409 kg.C	32 tonne.C

Technical Information	
Carbon Emission Index (kg.C/kWh)	0.117 kg.C/kWh

Appendices – Graphs

Figure 5.1.1a - Empirical Free Cooling % Vs Dry Bulb Outside Temperature Graph Based on ChW Flow / Return Temperature of 8 / 14 degC

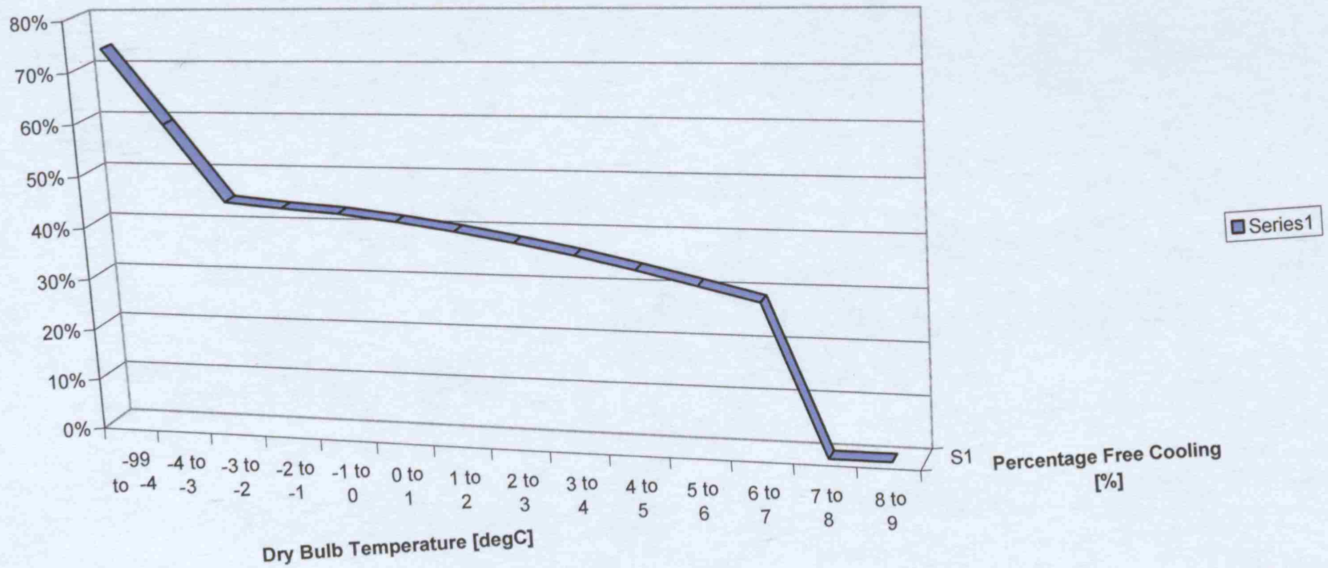


Figure 5.1.1b - Graph Showing Annual Total Free Cooling Vs Temperature Based on ChW Flow / Return Temperature of 8 / 14 degC (Run / Standby Arrangement)

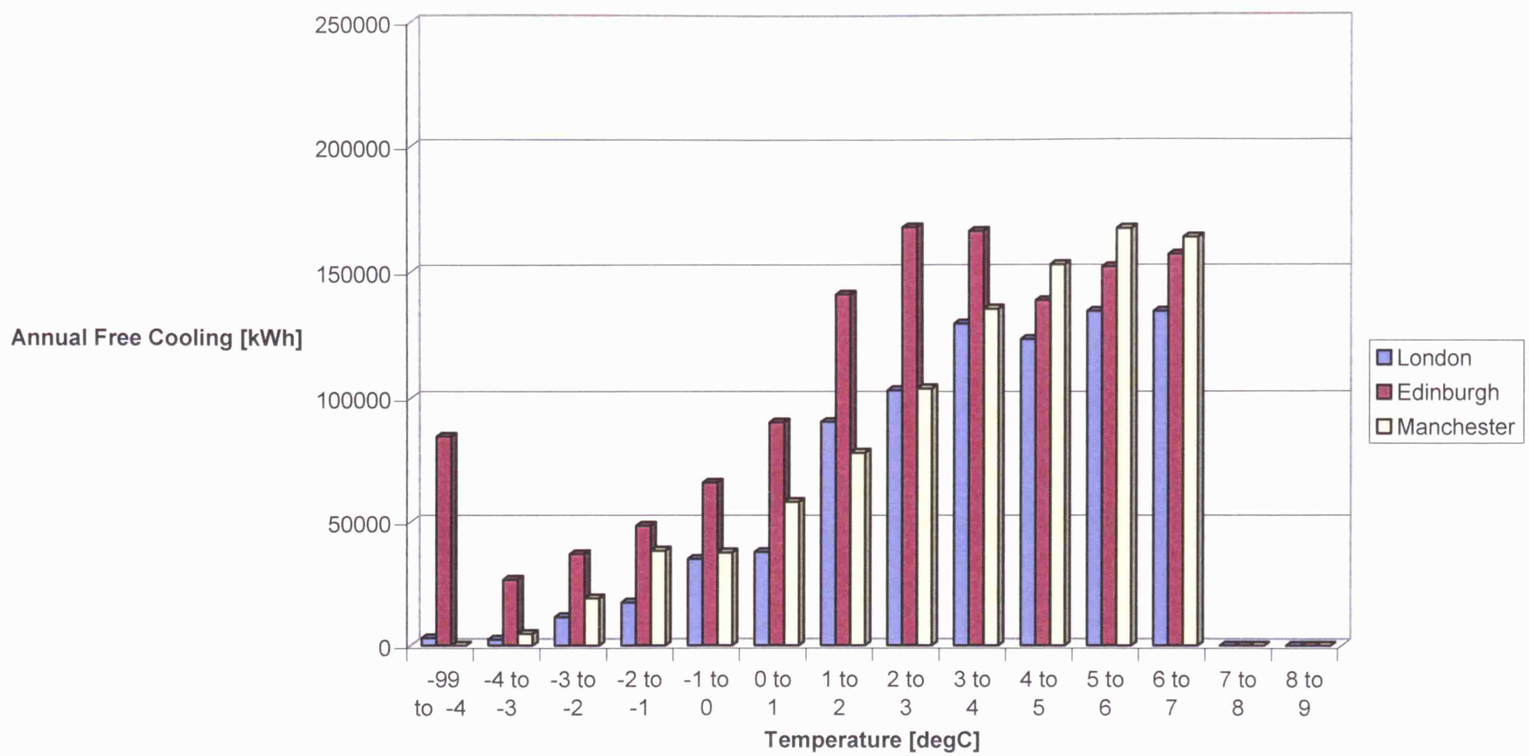


Figure 5.1.1c - Annual Total Cooling Required Vs Actual Free Cooling Graph Based on 4 No. Run / 4 No. Standby Chillers (ChW Flow / Return Temperature of 8 / 14 degC)

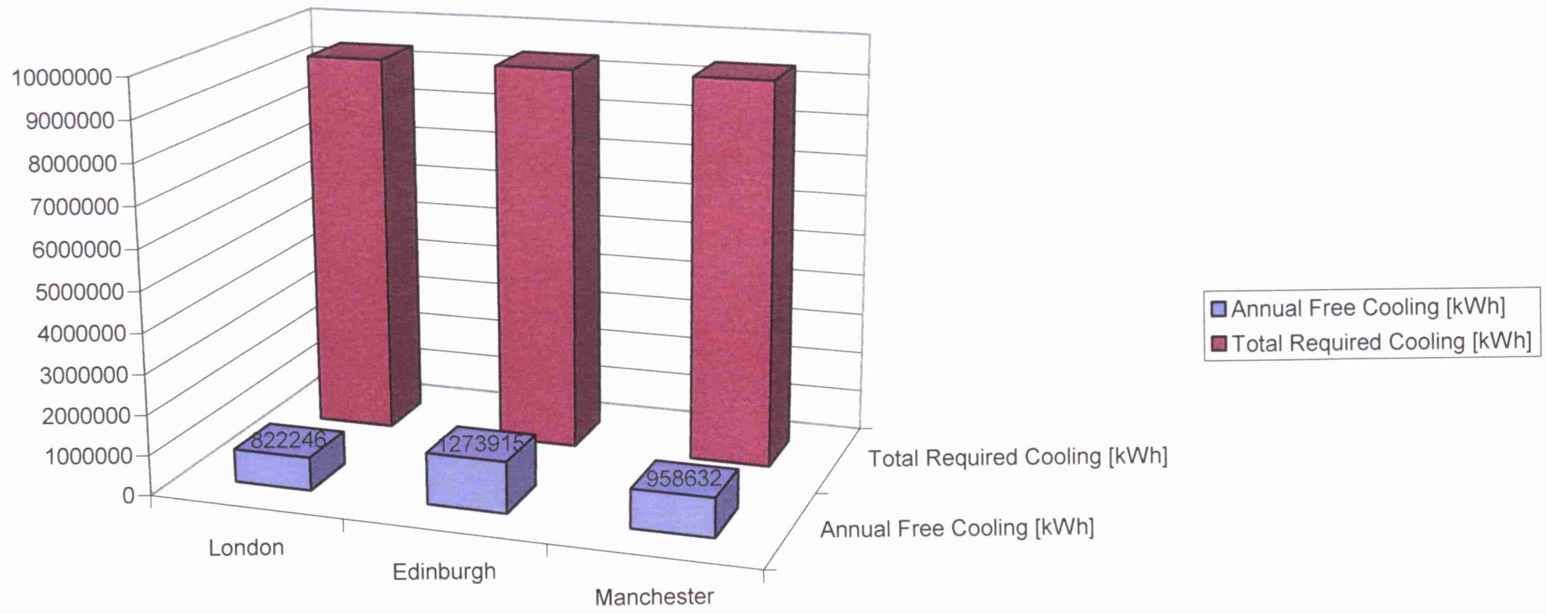


Figure 5.1.1d - Annual Total Cooling Required Vs Actual Free Cooling Graph Based on 8 No. Chillers in 'Hot Standby' Arrangement (ChW Flow / Return Temperature of 8 / 14 degC)

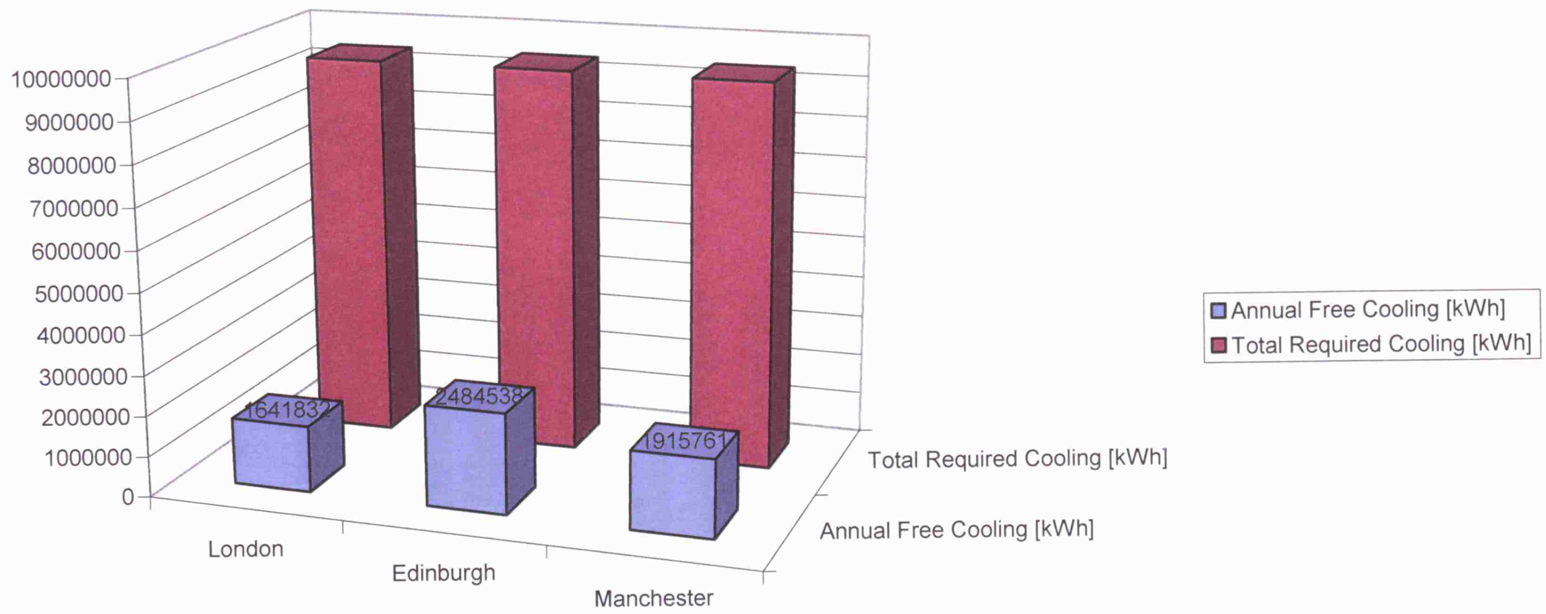


Figure 5.1.1e - Graph Showing Total Cooling Required Vs Available Free Cooling from Chillers running in 4 No. Run / 4 No. Standby Arrangement (ChW Flow / Return Temperature of 8 / 14 degC)

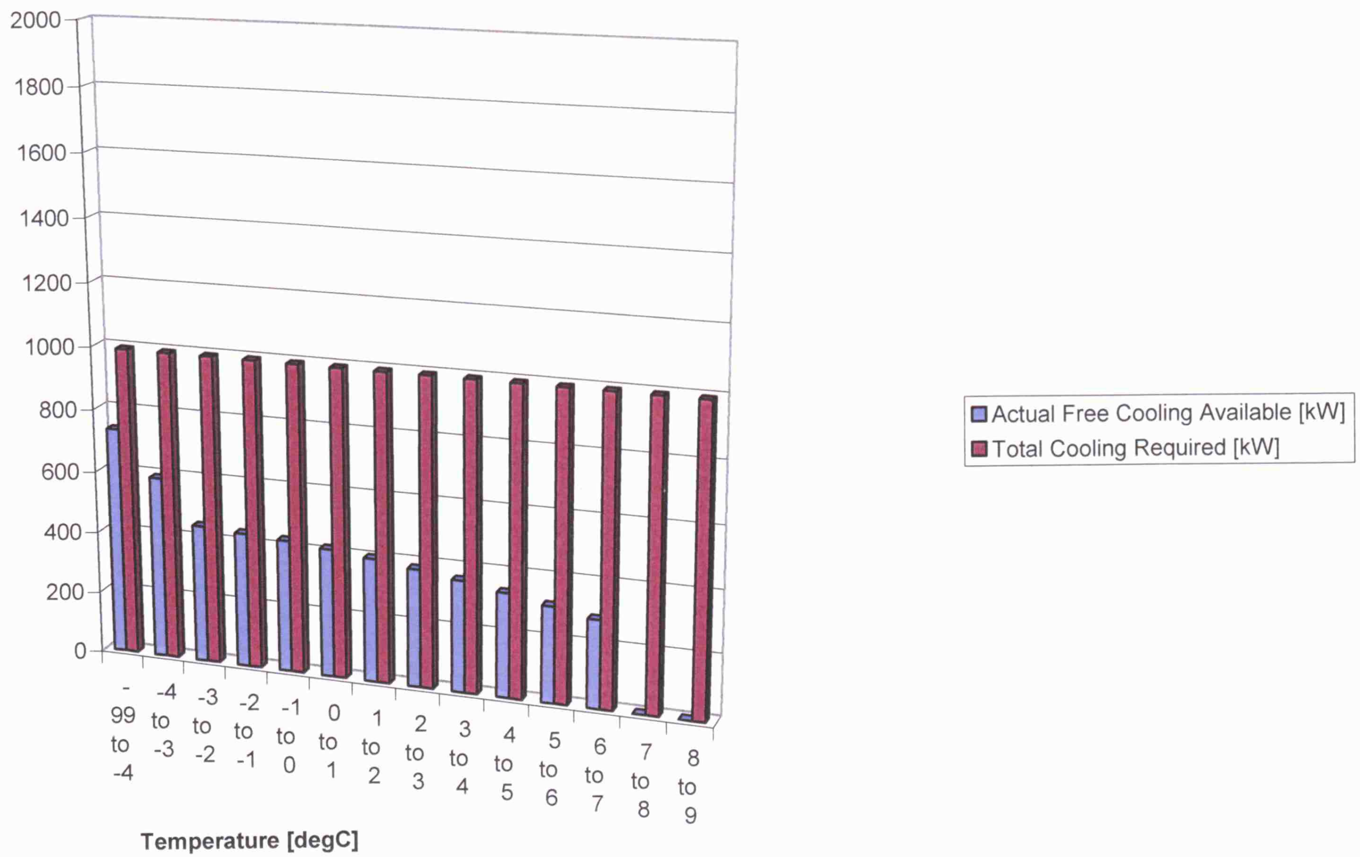


Figure 5.1.1f - Graph Showing Total Cooling Required Vs Available Free Cooling from 8 No. Chillers running in 'Hot Standby' Arrangement (ChW Flow / Return Temperature of 8 / 14 degC)

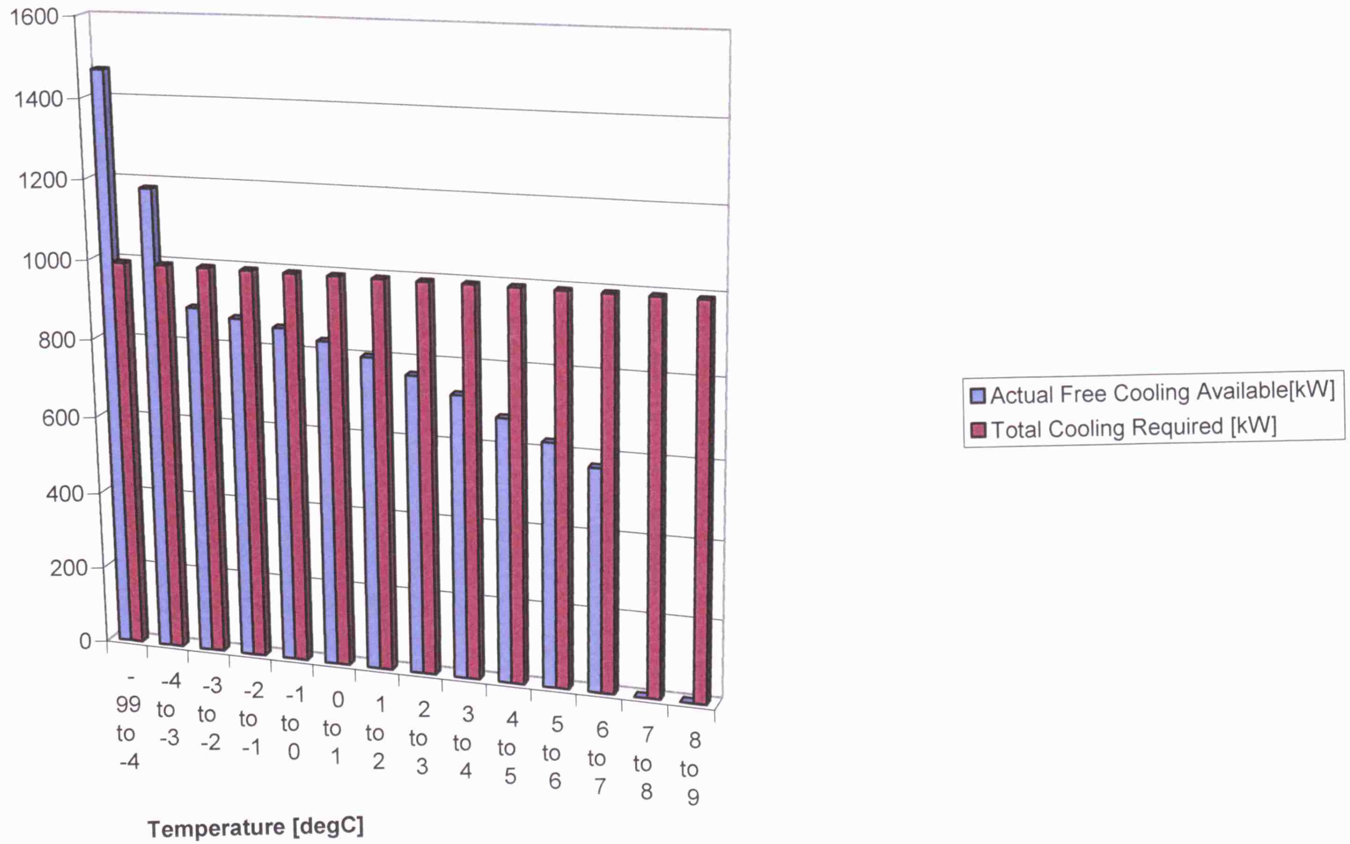


Figure 5.1.2a - Empirical Free Cooling % Vs Dry Bulb Outside Temperature Graph Based on ChW Flow / Return Temperature of 10 / 16 degC

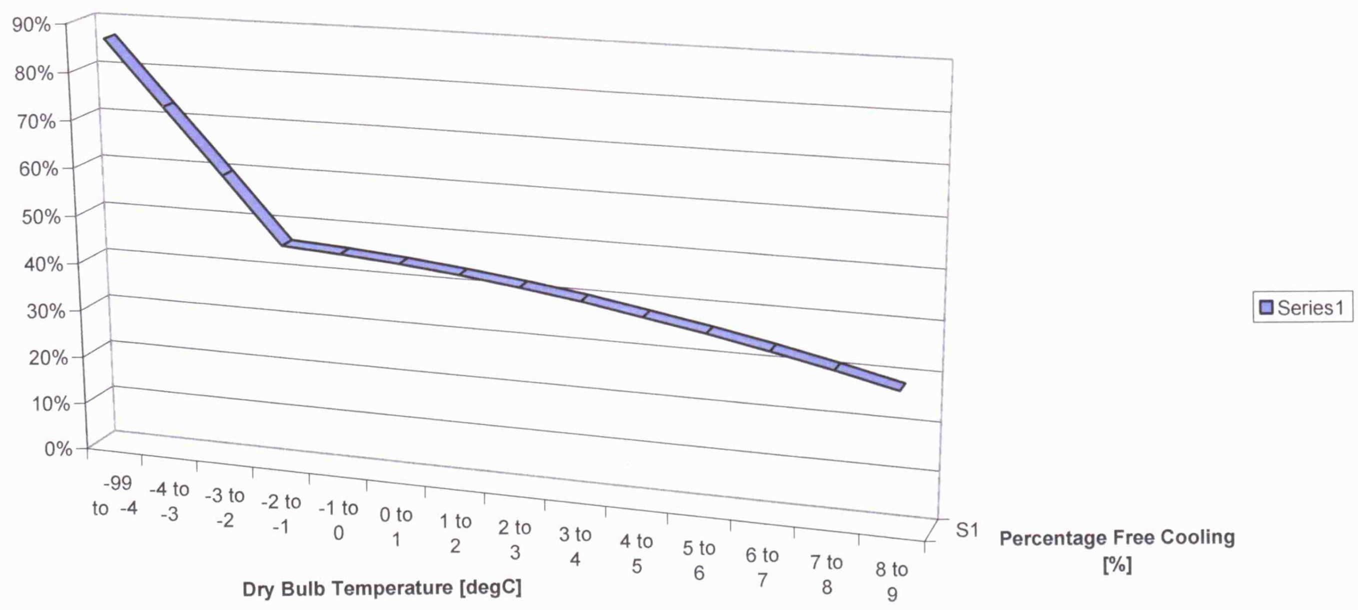


Figure 5.1.2b - Graph Showing Annual Total Free Cooling Vs Temperature Based on ChW Flow / Return Temperature of 10 / 16 degC

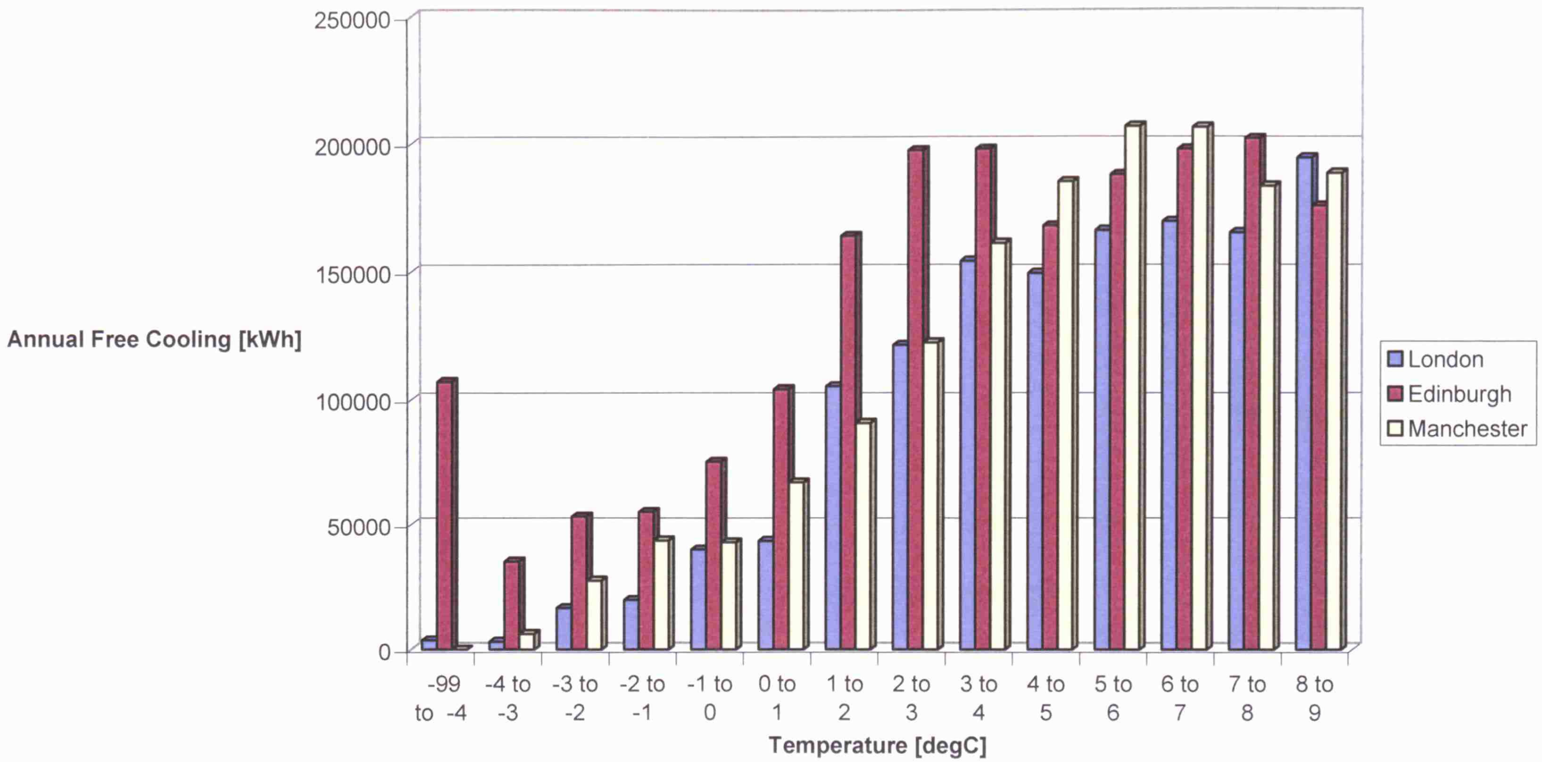


Figure 5.1.2c - Annual Total Cooling Required Vs Actual Free Cooling Graph Based on 4 No. Run / 4 No. Standby Chillers (ChW Flow / Return Temperature of 10 / 16 degC)

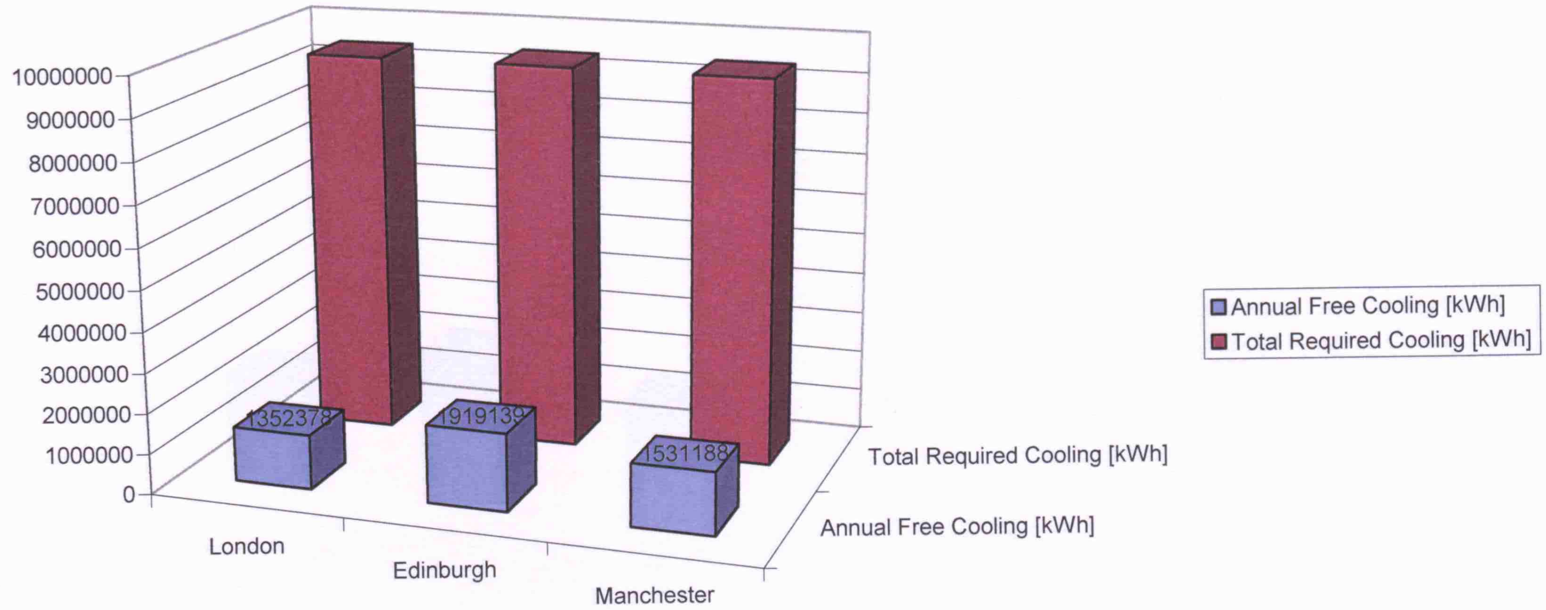


Figure 5.1.2d - Annual Total Cooling Required Vs Actual Free Cooling Graph Based on 8 No. Chillers in 'Hot Standby' Arrangement (ChW Flow / Return Temperature of 10 / 16 degC)

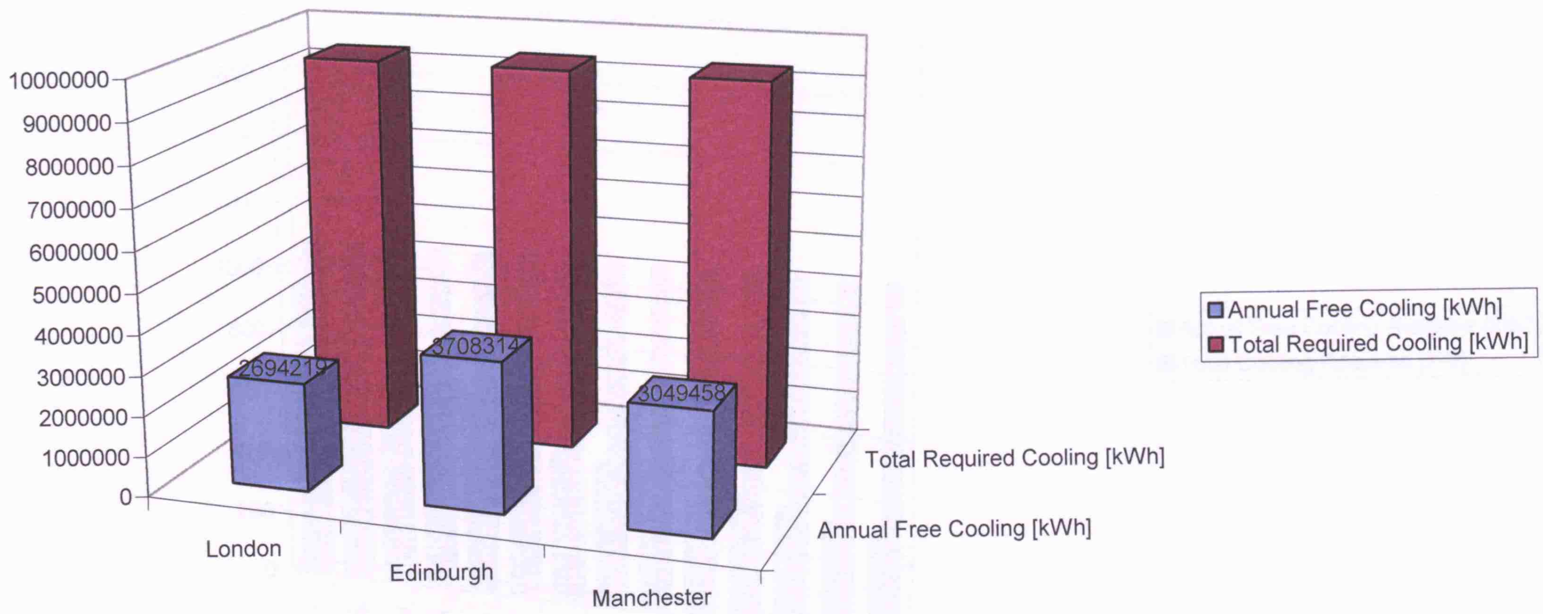


Figure 5.1.2e - Graph Showing Total Cooling Required Vs Available Free Cooling from Chillers running in 4 No. Run / 4 No. Standby Arrangement (ChW Flow / Return Temperature of 10 / 16 degC)

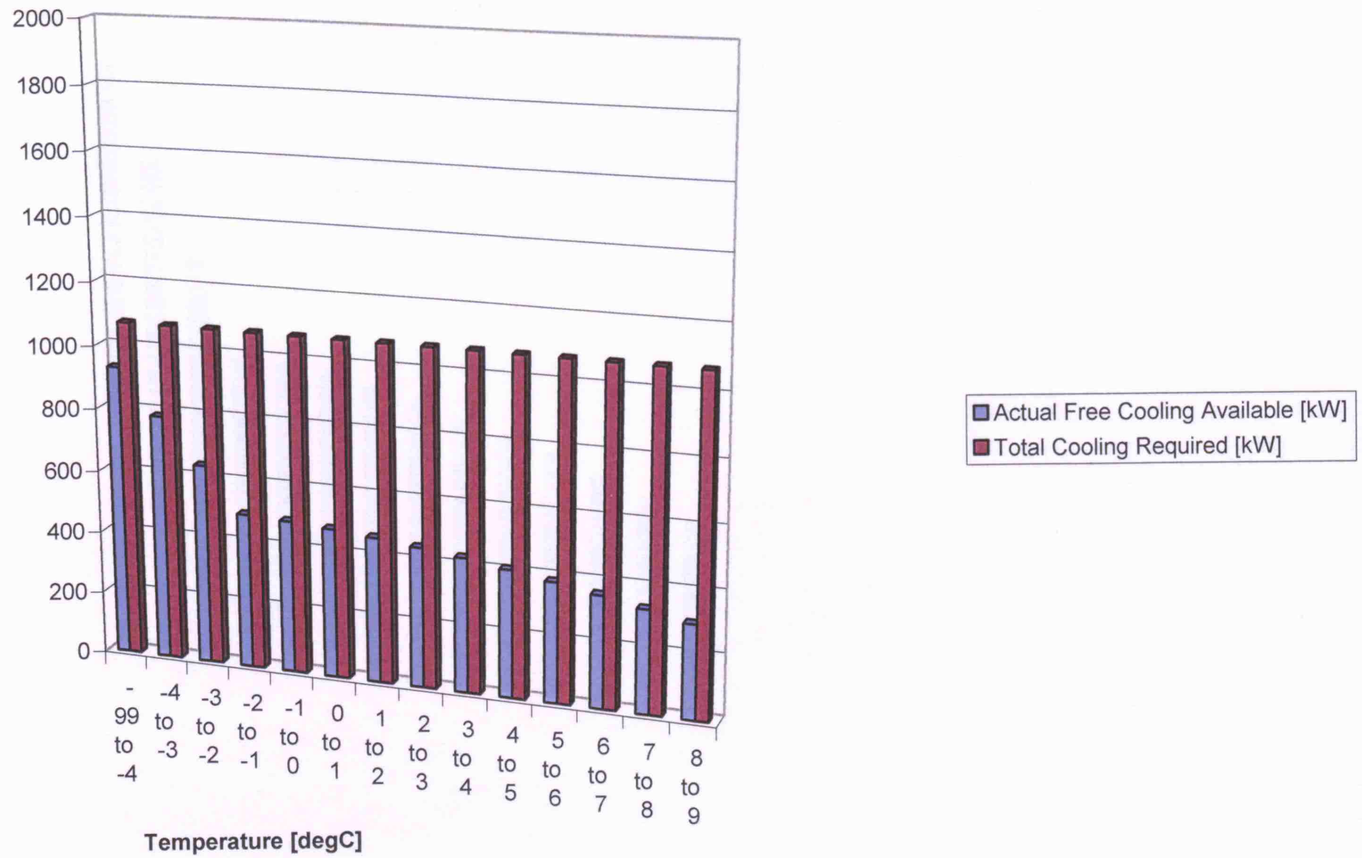


Figure 5.1.2f - Graph Showing Total Cooling Required Vs Available Free Cooling from 8 No. Chillers running in 'Hot Standby' Arrangement (ChW Flow / Return Temperature of 10 / 16 degC)

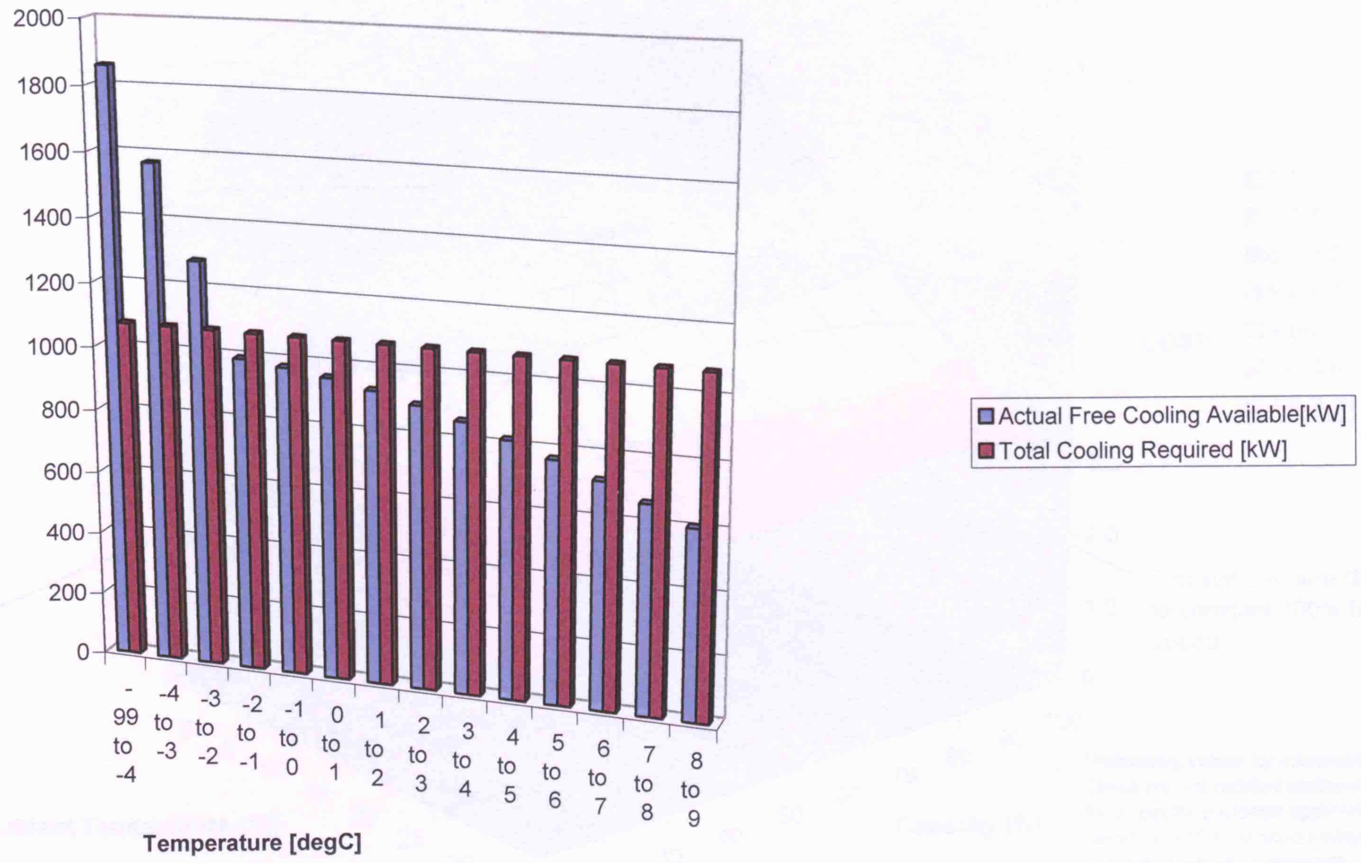
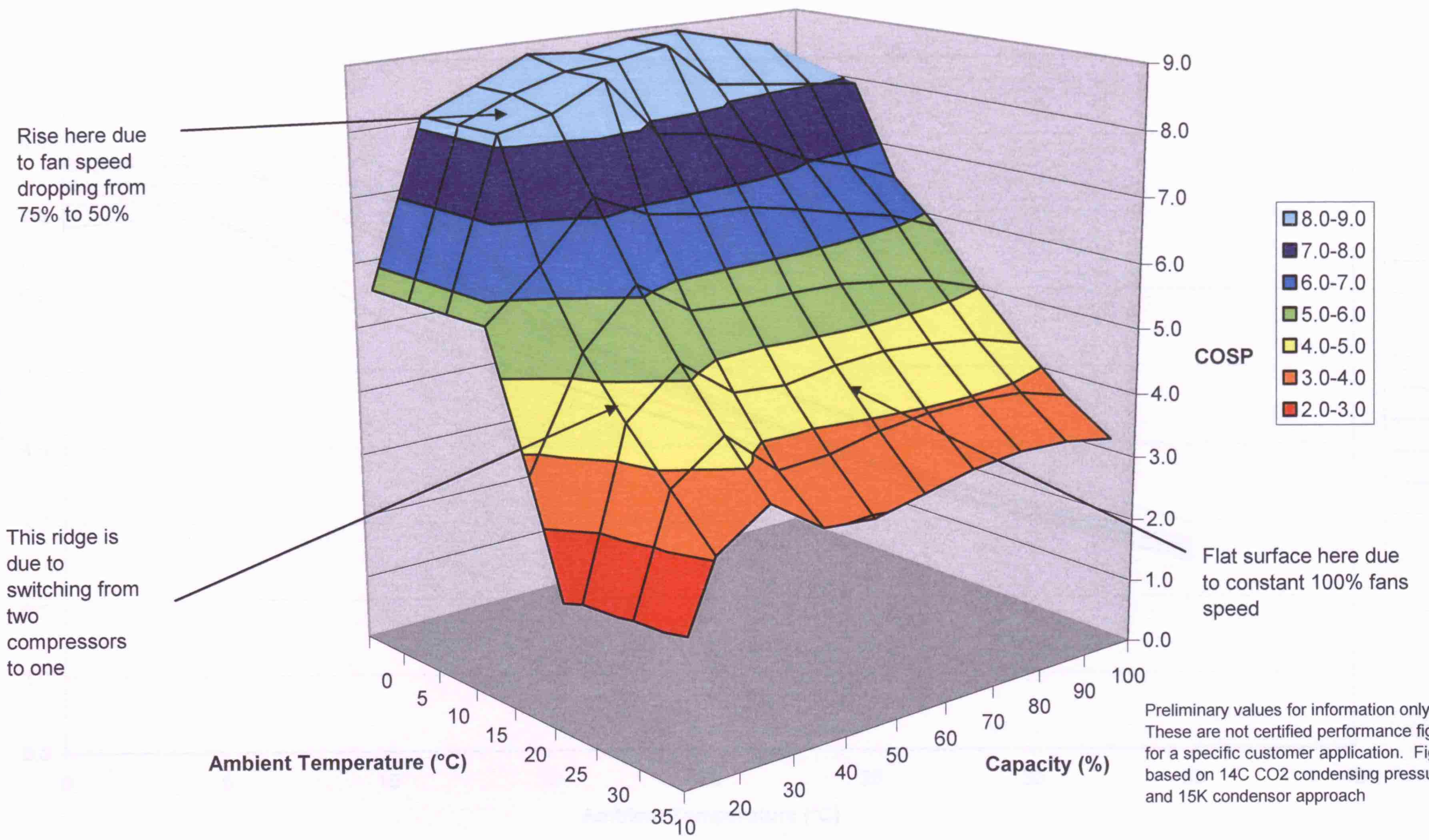


Figure 5.2.1a - Standard 500kW Air Cooled CO2OL Pack



Preliminary values for information only. These are not certified performance figures for a specific customer application. Figures based on 14C CO2 condensing pressure and 15K condensor approach

Figure 5.2.1b Standard 500kW Air Cooled CO2OL COSP Vs Ambient Temperature Graph (For Range Of Capacities)

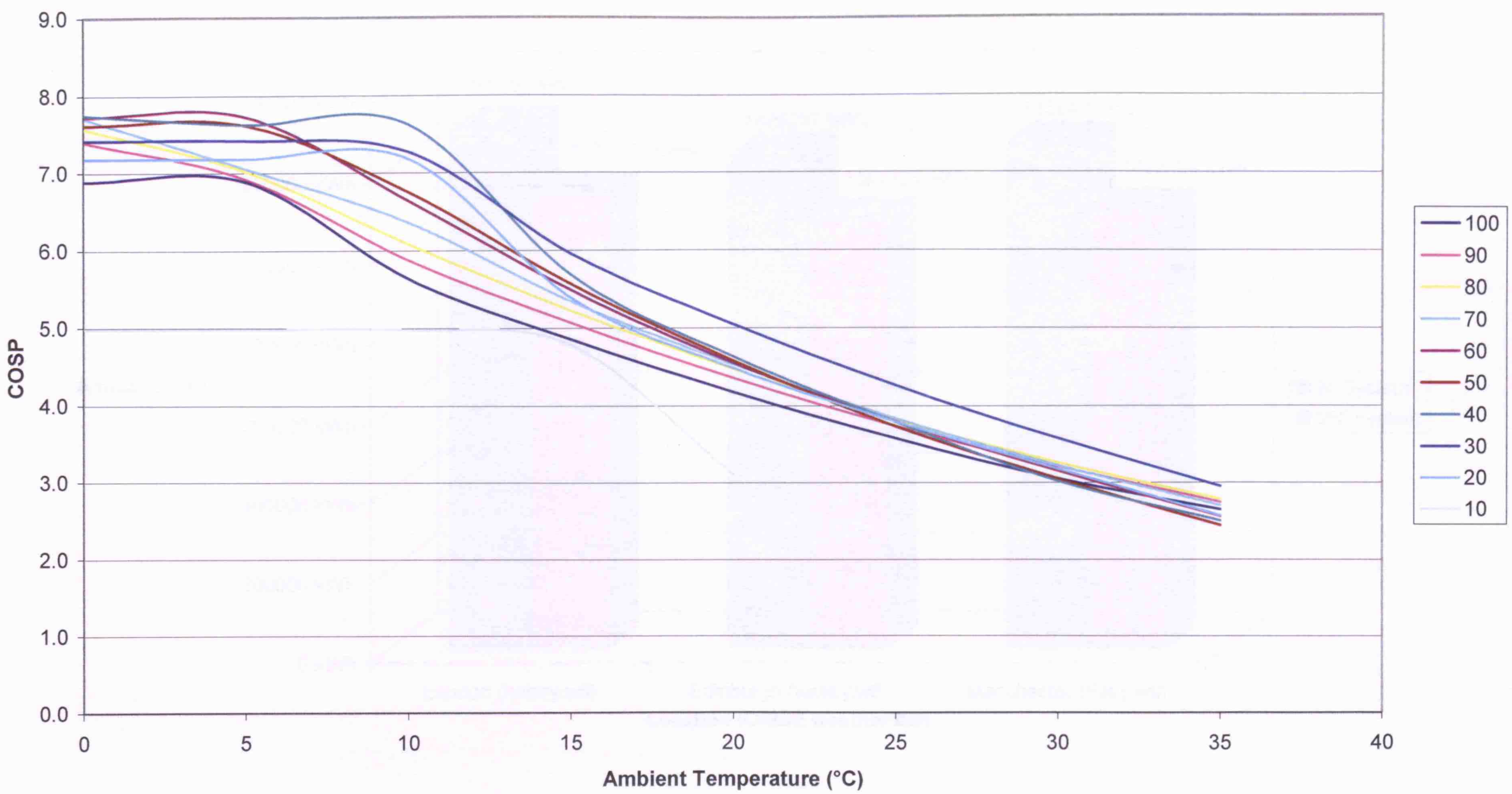


Figure 5.2.1c - Graph Showing Annual Energy Consumed Utilising Either a 'Run / Standby' Arrangement or a 'Hot Standby' Arrangement Based on Geographical Location

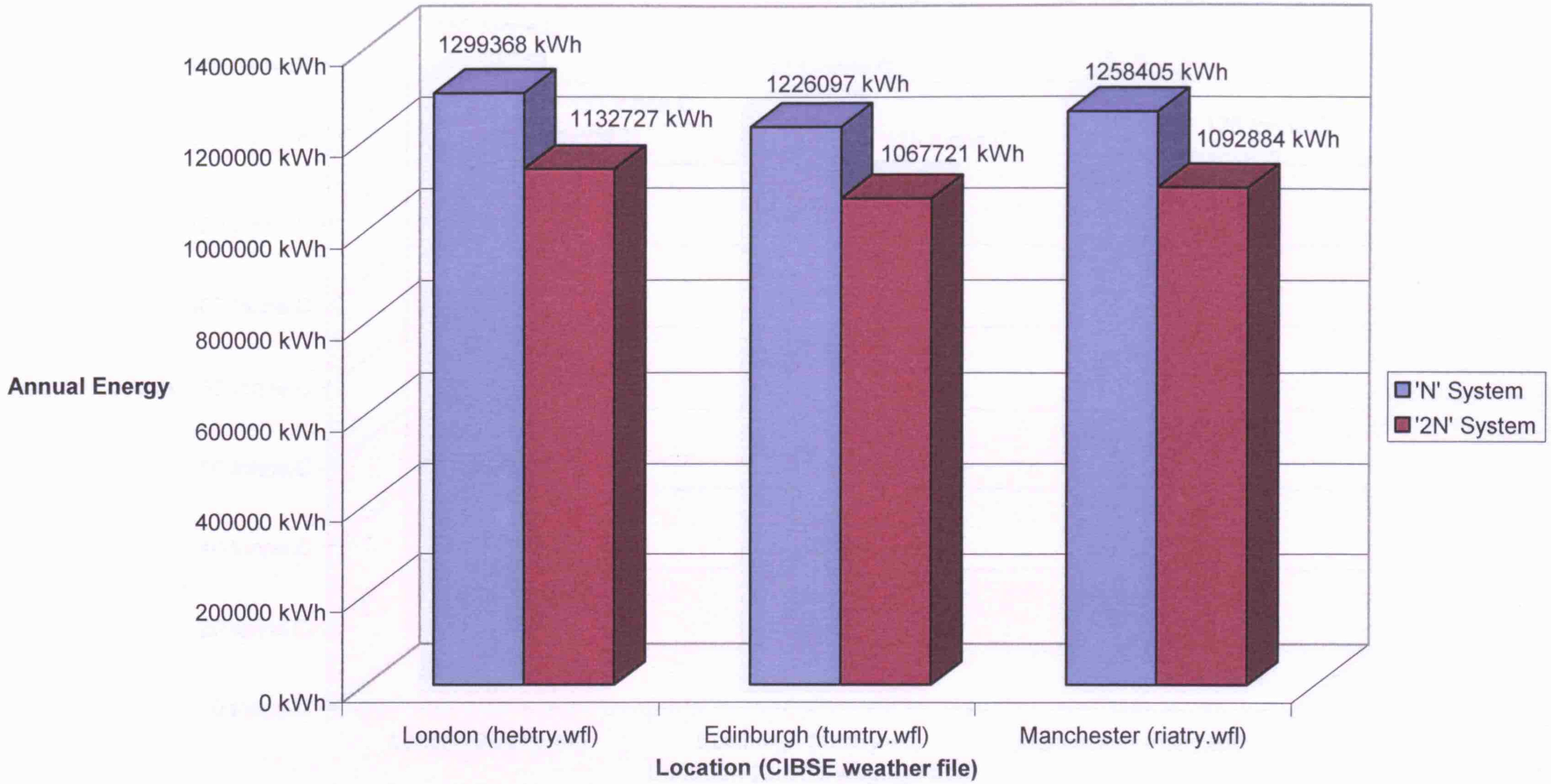


Figure 5.2.1d - Graph Showing Annual Carbon Produced Utilising Either a 'Run / Standby' Arrangement or a 'Hot Standby' Arrangement Based on Geographical Location

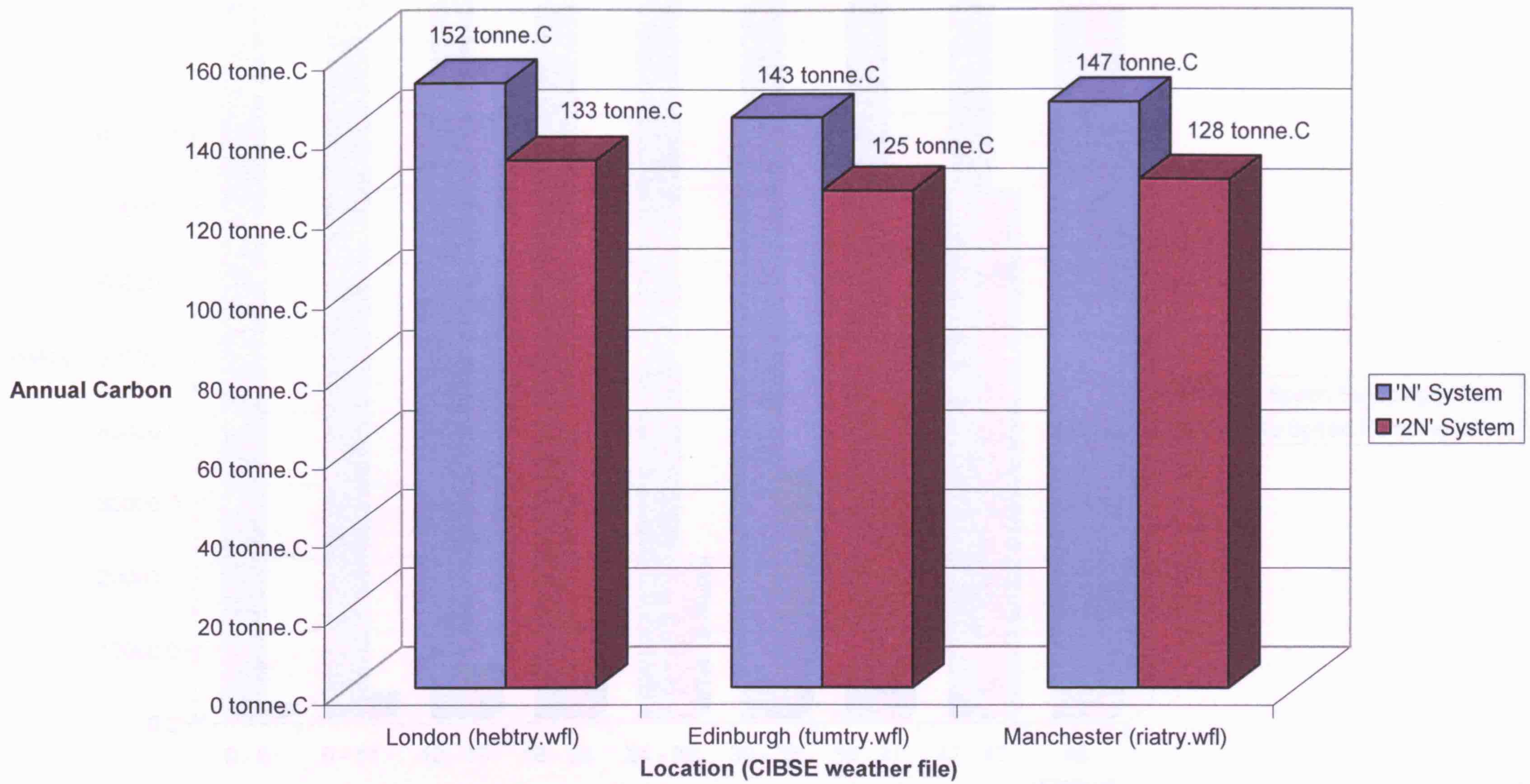


Figure 5.3.1a - Graph Showing Energy Consumption Comparison of Single Speed Vs Variable Speed Pumping for Stepped Population of Data Centre (0 - 48 Months)

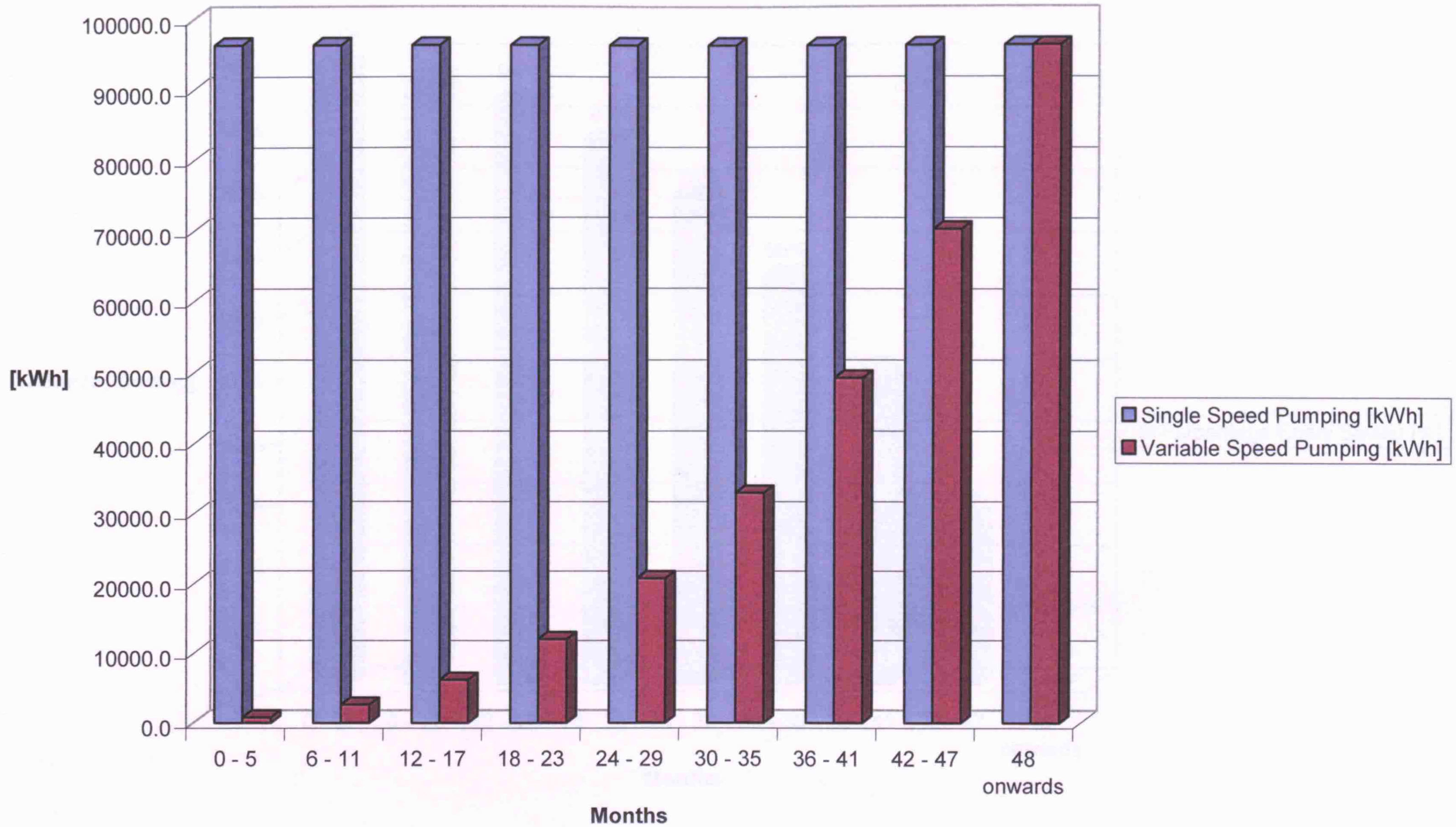


Figure 5.3.1b - Graph Showing Energy Consumption Percentage Saving Using Variable Speed Pumping (2 Port Control) Vs Single Speed Pumping (3 Port Control) Over 0 - 48 Months

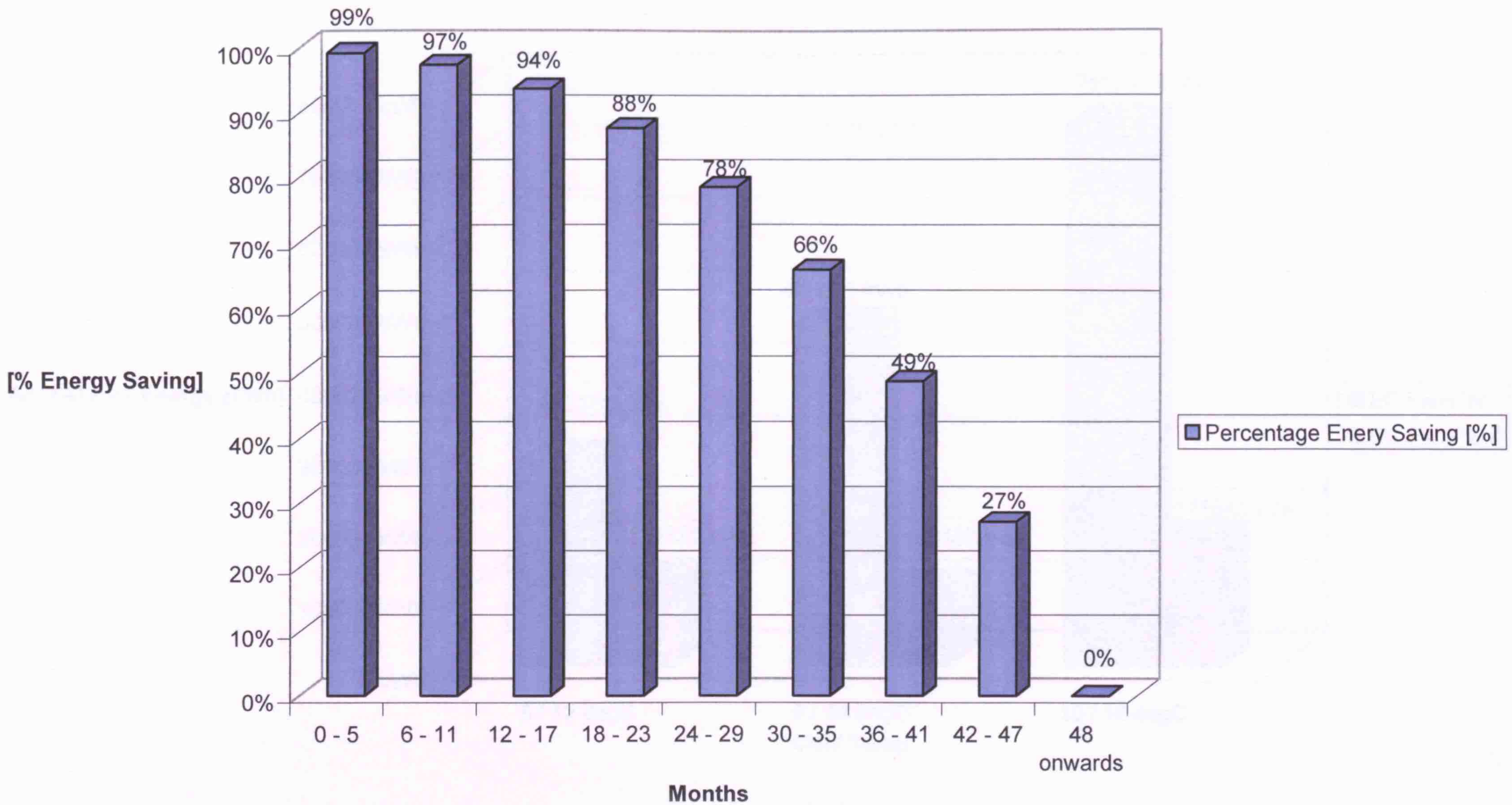


Figure 5.4.1a - Graph Showing Annual Energy Consumption for Electonically Commutated Fans (65 kW CRAC Unit) at Varying Water Temperatures in both 'N' and '2N' Running Arrangements

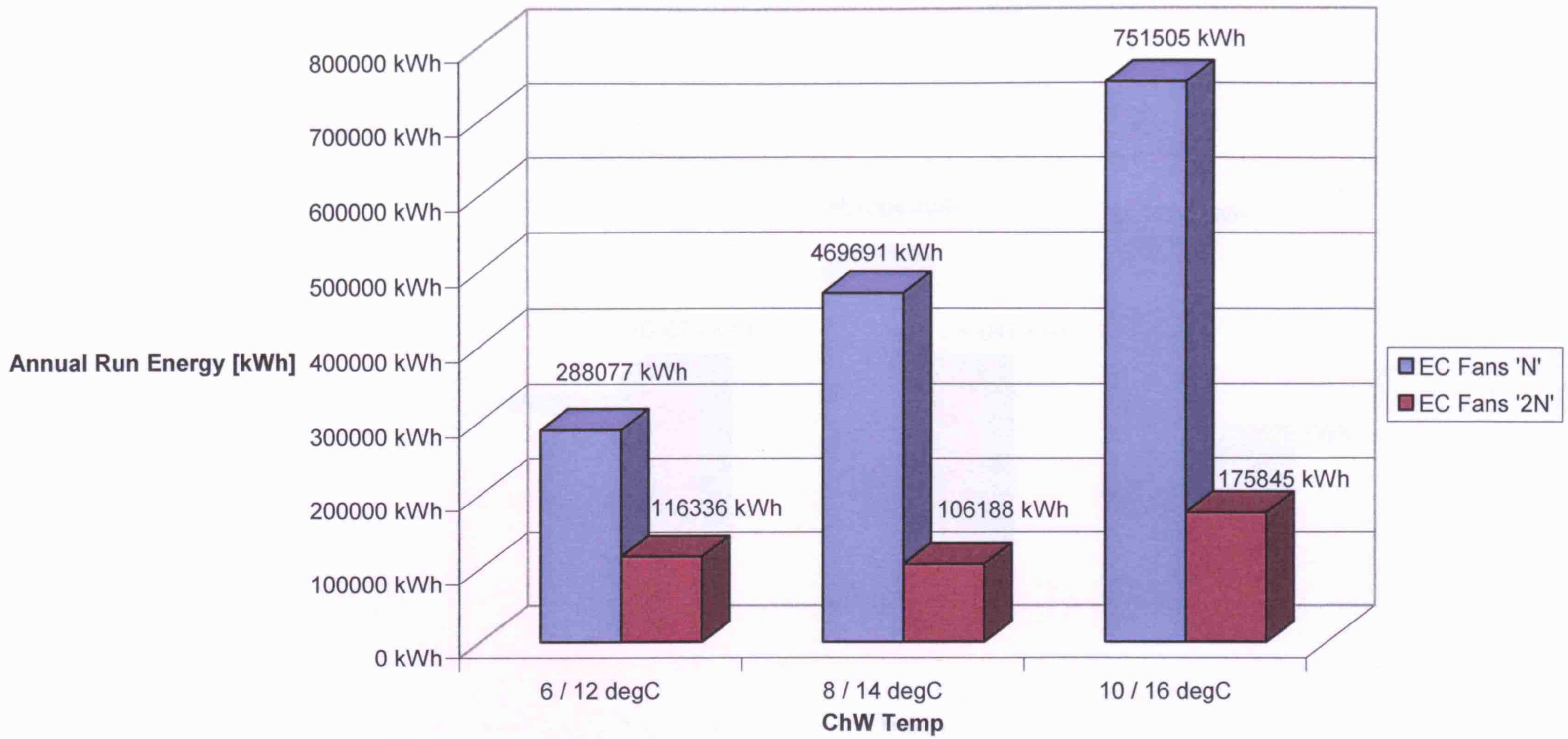


Figure 5.4.1b - Graph Showing Annual Energy Consumption for Centrifugal Fans (65 kW CRAC Unit) at Varying Water Temperatures in both 'N' and '2N' Running Arrangements

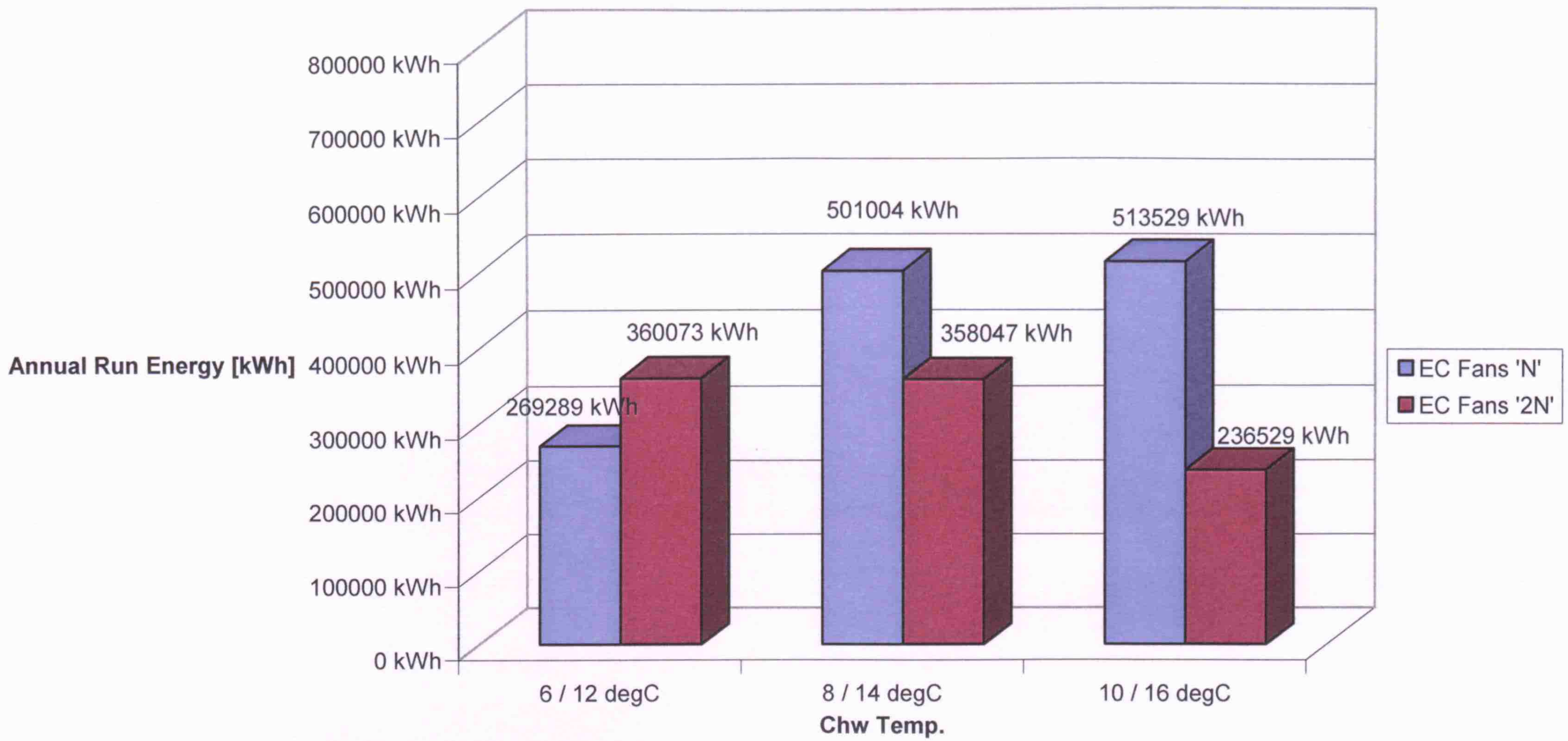
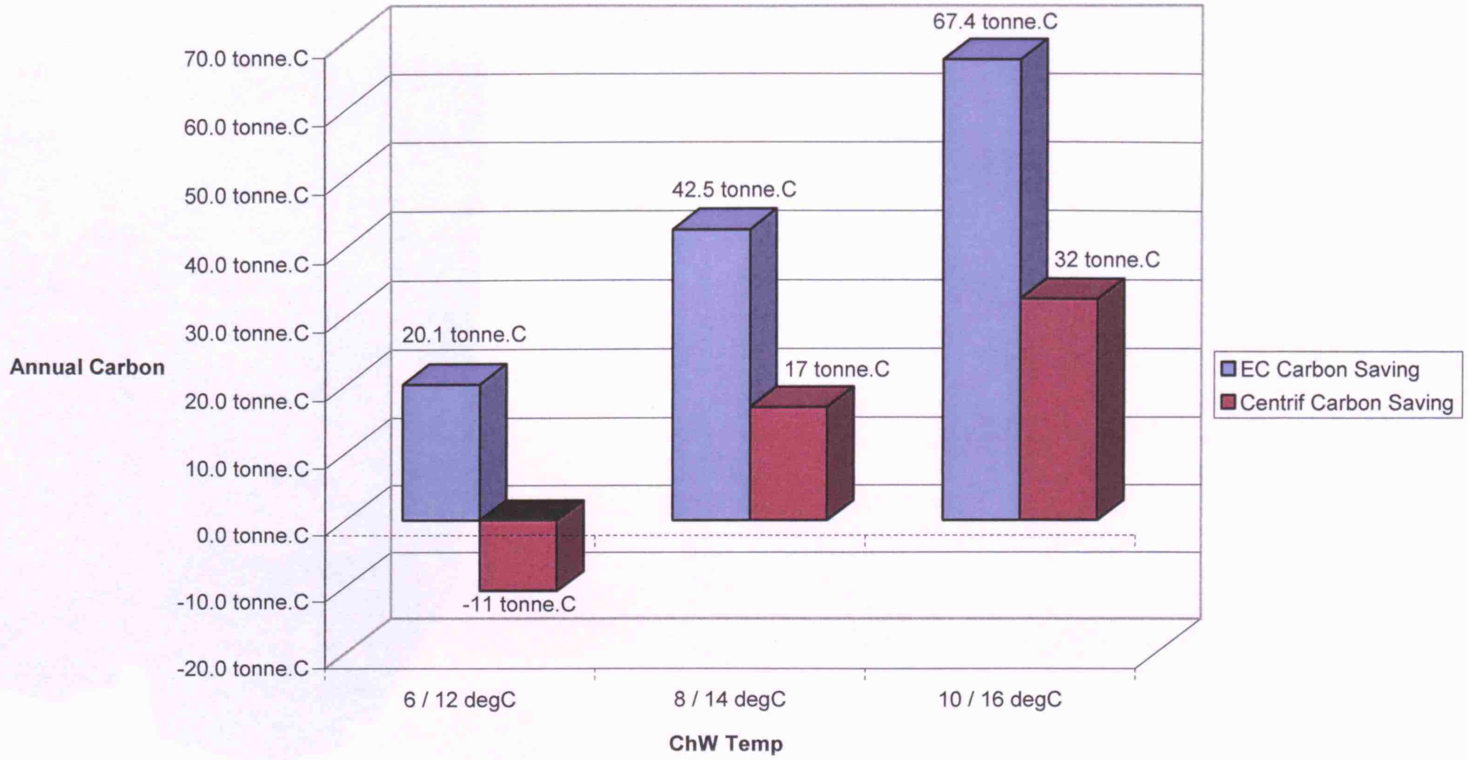
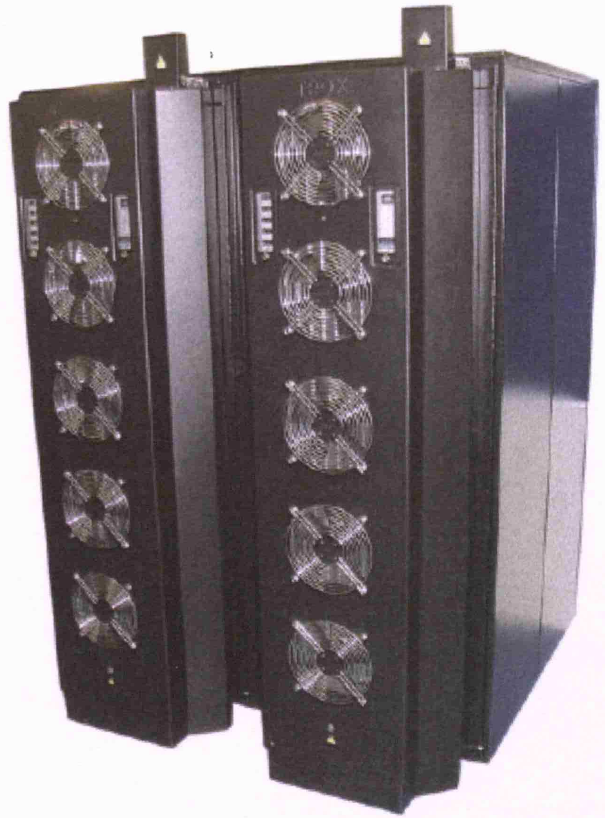


Figure 5.4.1c - Graph Showing Tonnes of Carbon Saving / Expense Based Upon Varying Chillled Water Temperature for both EC and Centrifugal Fans



Appendices – Photographs

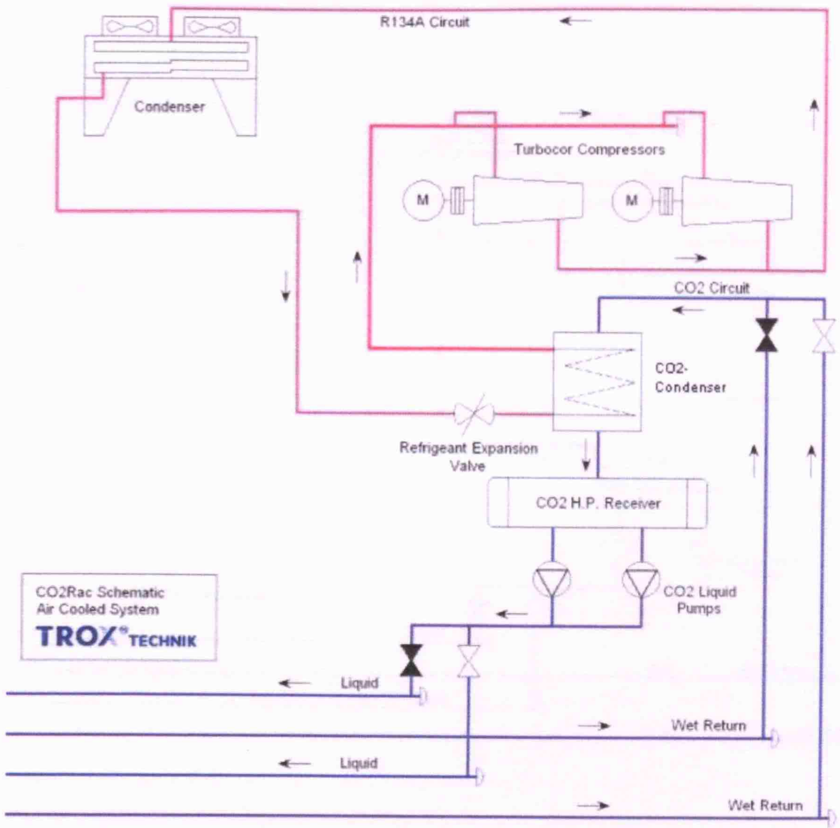
Photo 5.2.2a – Photo of Trox AITCS Indoor Rack Mounted Unit (Nominally 42U)



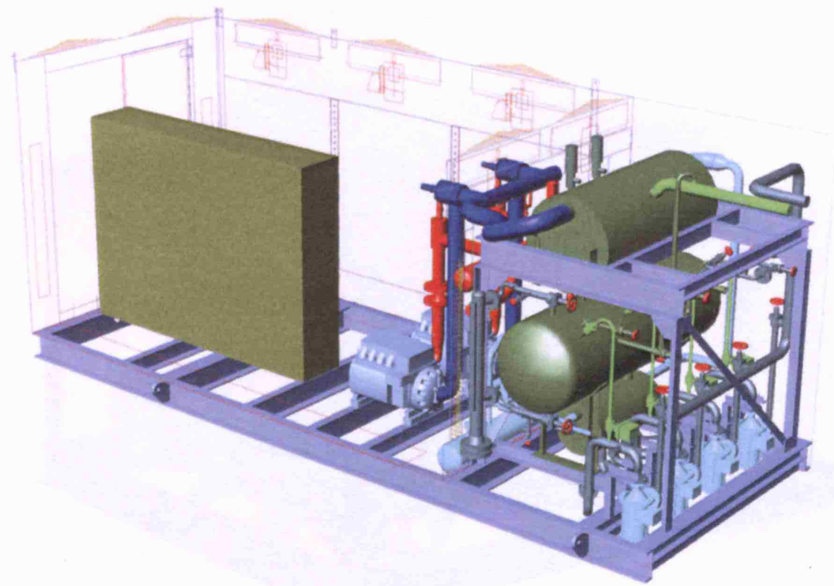
Appendices – Schematic Representations

See next page.

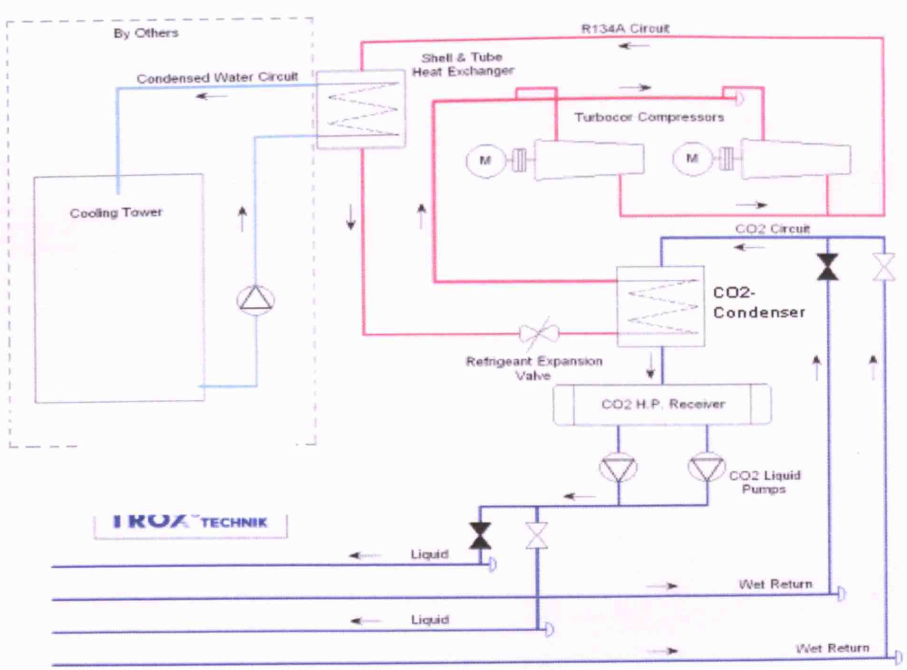
Schematic 5.2.2a – Schematic Representation of Trox AITCS Air Cooled System



Schematic 5.2.2b – Cutaway Representation of Trox AITCS Air Cooled System



Schematic 5.2.2c – Schematic Representation of Trox AITCS Water Cooled System



Schematic 5.2.2d – Cutaway Representation of Trox AITCS Water Cooled System